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USAAVLABS TECHNICAL REPORT 64-30

EVALUATION STUDY
OF
BALANCED LOAD PLANETARY TRANSMISSION
AND
DESIGN OF 250-HORSEPOWER UNIT
WITH
4000-HOUR TIME BETWEEN OVERHAUL (TBO)

By

S. W. Baker
C. S. Davis, Jr.
W. H. Schwab

January 1966

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-102(T)

BERGEN RESEARCH ENGINEERING CORP.
TETERBORO, NEW JERSEY

Disclaimers

Disposition Instructions

ERRATUM

USAAVLABS Technical Report 64-30

January 1966

Page iii - Delete last paragraph.

DEPARTMENT OF THE ARMY
U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA 23604

This report covers the results of a segment of this command's program to study improved Army aircraft power transmission systems. Instant report specifically discusses the results of a parametric study and design of a balanced load planetary transmission concept.

The objective of this study was to evaluate the balanced load planetary system concept for compatibility with the high ratio requirements of helicopter transmissions utilizing gas turbine engines and to present the potential effectiveness in terms of reliability, weight reduction, and reduced complexity. The analysis was carried forward in the design of a 250-horsepower transmission which utilized this concept and was designed for time between overhaul of 4000 hours.

This command concurs with the conclusions of the contractor.

Task 1D121401D14404
Contract DA 44-177-AMC-102(T)
USAAVLABS Technical Report 64-30
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(Final Report 8314-R3)

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Prepared by
BERGEN RESEARCH ENGINEERING CORP.
Teterboro, New Jersey

for
U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

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SUMMARY

This report presents a complete review of the parametric design and computer studies used to evaluate the Bergen Research Compound Planetary Transmission concept in three configurations for a helicopter transmission as schematically illustrated in Figure 1. The concept studies all embody design arrangements which assure uniform load division with multiple compound planetary gears. Uniform loading is accomplished by balancing the primary and secondary meshing point positions through low rate springing. These systems stabilize at the gear mesh positions of maximum load path equalization.

Design requirements are established for typical horsepower ranges using consistent gear and bearing design factors. Design factors are studied at both 1000-hour and 4000-hour Time Before Overhaul (TBO) to determine feasibility and design trade-off requirements. Comparison of these includes design sizing of each type at 1500-horsepower and comparable life factors for evaluation of their relative merits and choice of the preferable configuration.

A mathematical model is developed to show the interrelationship of key design factors. A computer program provides a broad range of these parametric variations. The study results are applied in Phase II to the preferred design concept through completion of the layout and detail engineering. This design is suitable for fabrication and life testing.

Recommendations are included for establishment of programs to further investigate this type of transmission and the method of parametric study.

FOREWORD

The U. S. Army Aviation Materiel Laboratories entered into a contract for the purpose of performing a design study for an improved mechanical transmission system. This contract was designated as DA 44-177-AMC-102 (T).

The contract was divided into phases. Phase I, Parametric Study, included an analytical study of the Bergen Compound Planetary Transmission System* to determine the most advantageous configuration for application to current and projected Army aircraft. In Phase II, a detail design of a test unit of the selected design concept was made.

Work under the contract required the combined efforts of many Bergen Research Engineering Corporation personnel. The Entire program was conducted under the direction of Mr. S.W. Baker, President, Mr. C.S. Davis, Jr., Manager Advance Engineering, and Mr. W.H. Schwab, Chief Project Engineer. Significant contributions were made by Mr. W.G. Atkinson, Project Engineer, and Mr. F.W. Schwab, Contract Manager.

Bergen Research recognizes the effective liaison which existed between Mr. R.P. McKinnon and the contractor as being of great importance in the completion of this contract. The USAAVLABS project personnel, Mr. W. Hudgins and his branch chief, Mr. L. Bartone, provided a working relationship of the highest quality.

*Patent No. 3,144,790 and Application No. 261038

CONTENTS

ILLUSTRATIONS

SYMBOLS

		Page
b	width of Hertzian contact - inches	26
C	roller bearing capacity	27
C	cost of procurement & maintenance per year	51
d	roller diameter - inches	27
D	pitch diameter - inches	27
D	bearing diameter - inches	54
F	active face width - inches	26
G	number of trunnions	29
G	number of years per generation	51
H	hardness - BHN	25
i	number of rows per bearing	27
K	approximate constant	29
l	length of one roller - inches	27
L	design life - hours	28
m	ratio	29
M	merit factor	29
n	number of cycles	25
n	number of overhauls per generation	51
N	number of teeth	27
N	number of helicopters	51
P	pinion bearing load	55
Q	input torque - inch pounds	27

SUBSCRIPTS

a	ambient
b	bearing
b	Brinell
c	lowest point of contact
g	gear set
i	input
o	output
r	roller
1	sun gear
2	ring gear
3	primary planet
4	secondary planet

SYMBOLS

PARAMETRIC STUDY RESULTS

The results of the parametric study of the three types of planetary transmissions (Figure 1) indicate that all three are exceptionally light and efficient. It is noted however, that Types I and III are much lighter than Type II. Type II is heavier because twelve narrow secondary gear mesh widths cause inefficient weight design. If experience should prove actual whiffletree load carrying capacity twice that currently assumed, then the gear widths (which are unusually narrow) can be doubled and the design with proper bearings would have twice the torque rating at considerably less than twice the weight. This will occur since the weight of gear webs, pinion shafting, bearing housings, et cetera, would not change linearly with gear width. In that event, the Type II with six pinions and the whiffletree dual input sun gear arrangements would probably be the lightest design. For the present, however, active consideration of Type II is deferred in favor of Types I and III. The results of comparing Type I and Type III have been plotted against various indices of work capacity (Figures 2 and 3). These graphs show that the transmission weight of Types I and III are approximately the same at .27 to .50 pounds/horsepower and 1.15 to 1.9 pounds/1000 inch-pounds rotor torque. The increase in the pounds per horsepower figure at the higher horsepower is due to the decrease in the output rotor RPM with corresponding increase in output torque. The weight difference between 4000-hour and 1000-hour designs is of the order of 10 to 15 percent. The designs were all calculated with 4000-hour gearing so the weight differences arise from the necessity to accommodate more bearing capacity. It is significant that both designs are fairly similar in weight and both are significantly lighter than Type II. At the 460,000 inch/pound, 1500-horsepower design point, the most accurate comparison was made. The difference here favors the Type III by about 3 percent but more careful detailed design of each might reverse or neutralize the small difference. The conclusion is, therefore, that both designs are lightweight and the choice between them should be based upon other major characteristics.

Transmission size is controlled mainly by the required diameter for an adequate gear train. Figures 4, 5, and 6, indicate the variation of diameter, frontal area, and volume versus horsepower. The size is well within the usual requirements fitting easily into airframe and rotor proportions. The Type I design at 4000-hour design life requires somewhat larger diameter than at 1000-hour life because the

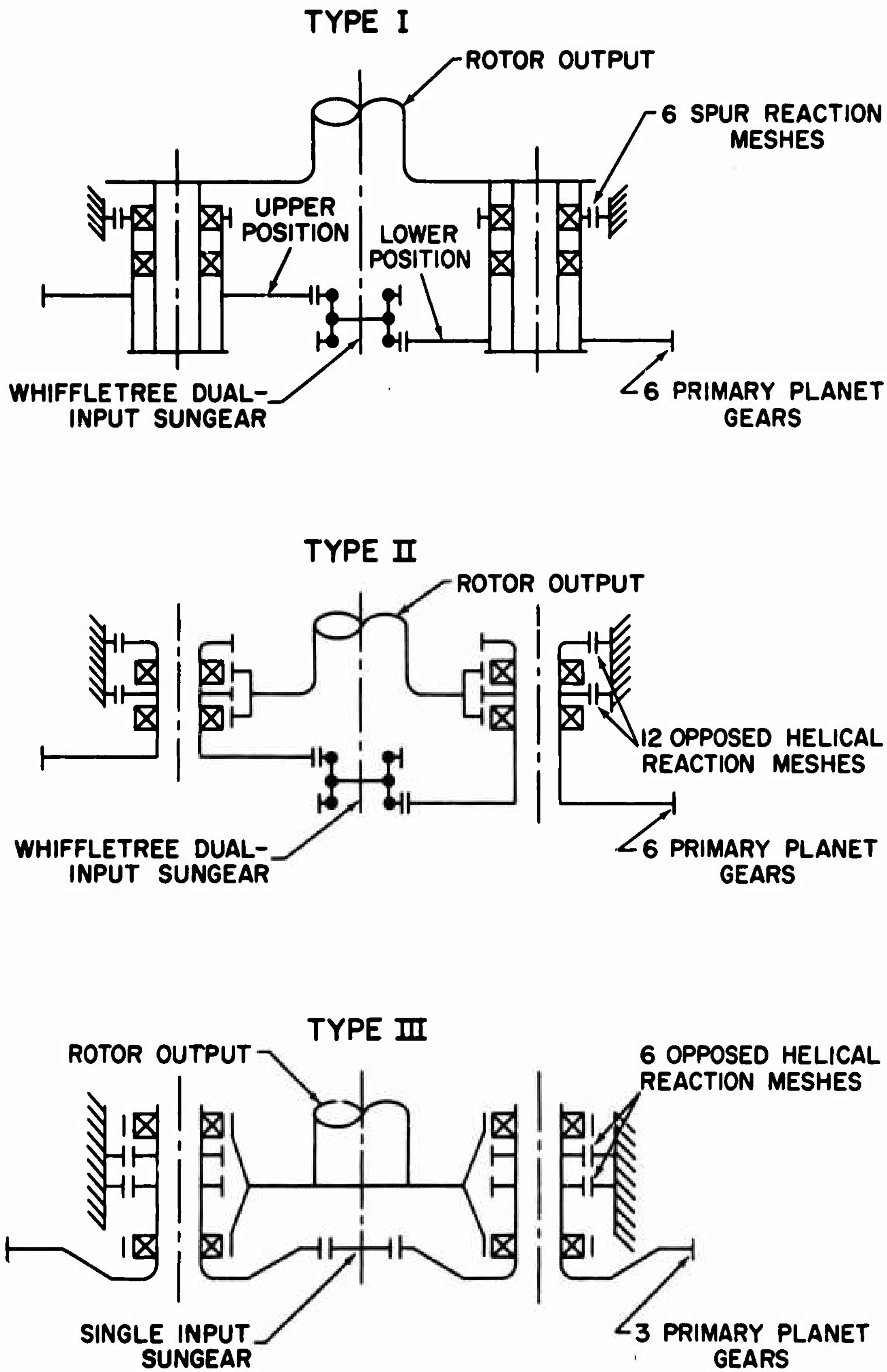


Figure 1. Transmission Configurations

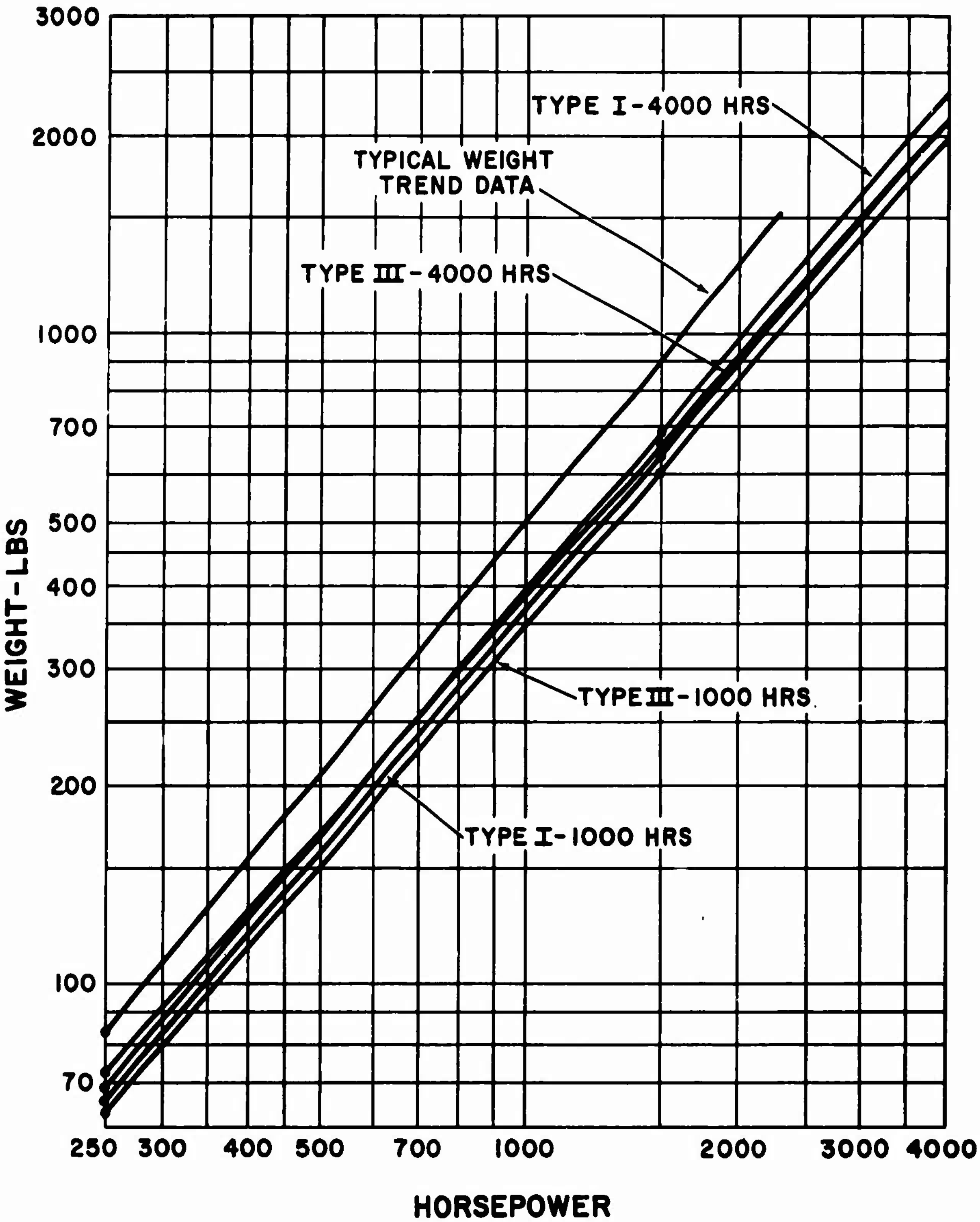


Figure 2. Transmission Weight Vs. Horsepower for Varying Transmission Life

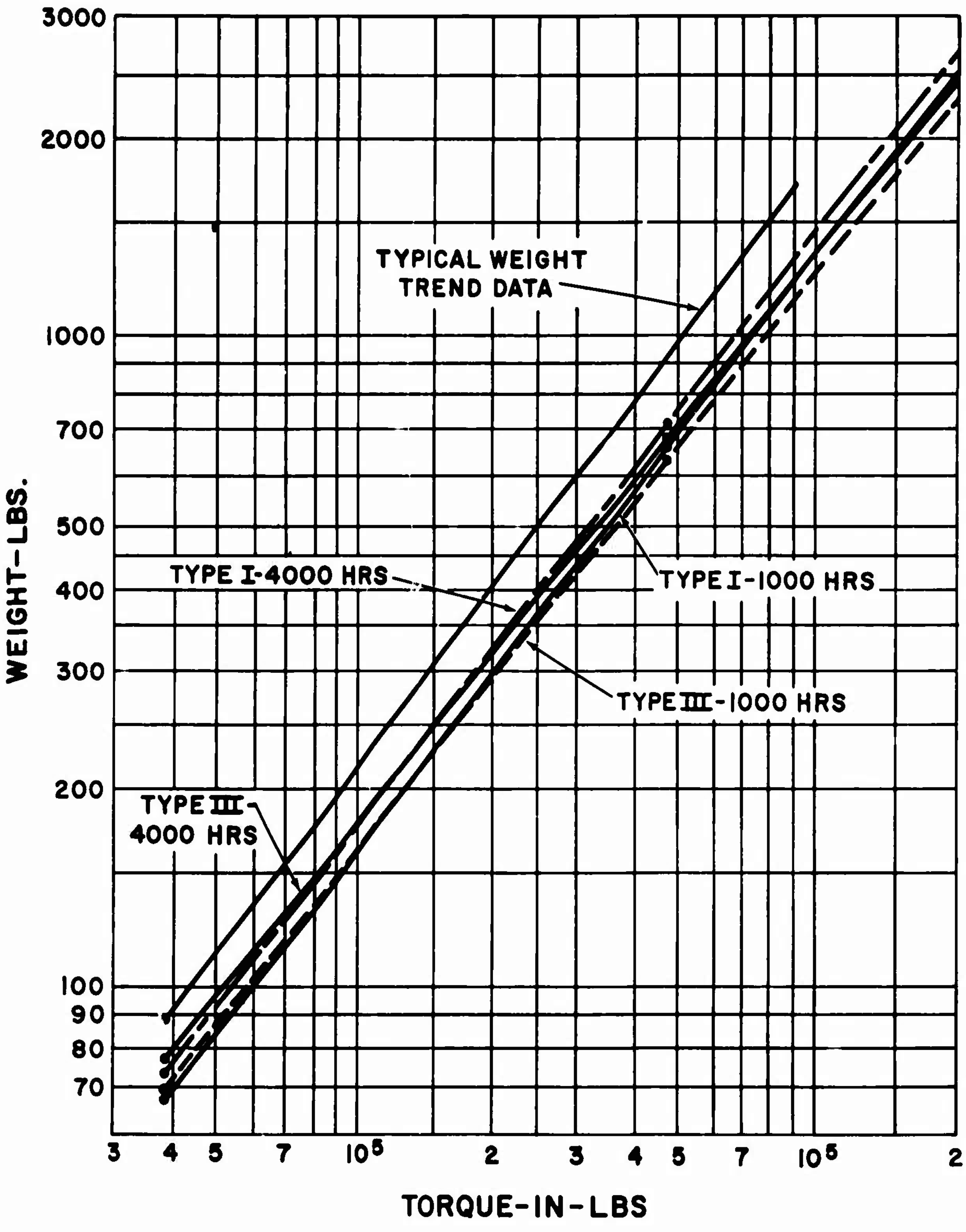


Figure 3. Transmission Weight Vs. Torque for Varying Transmission Life

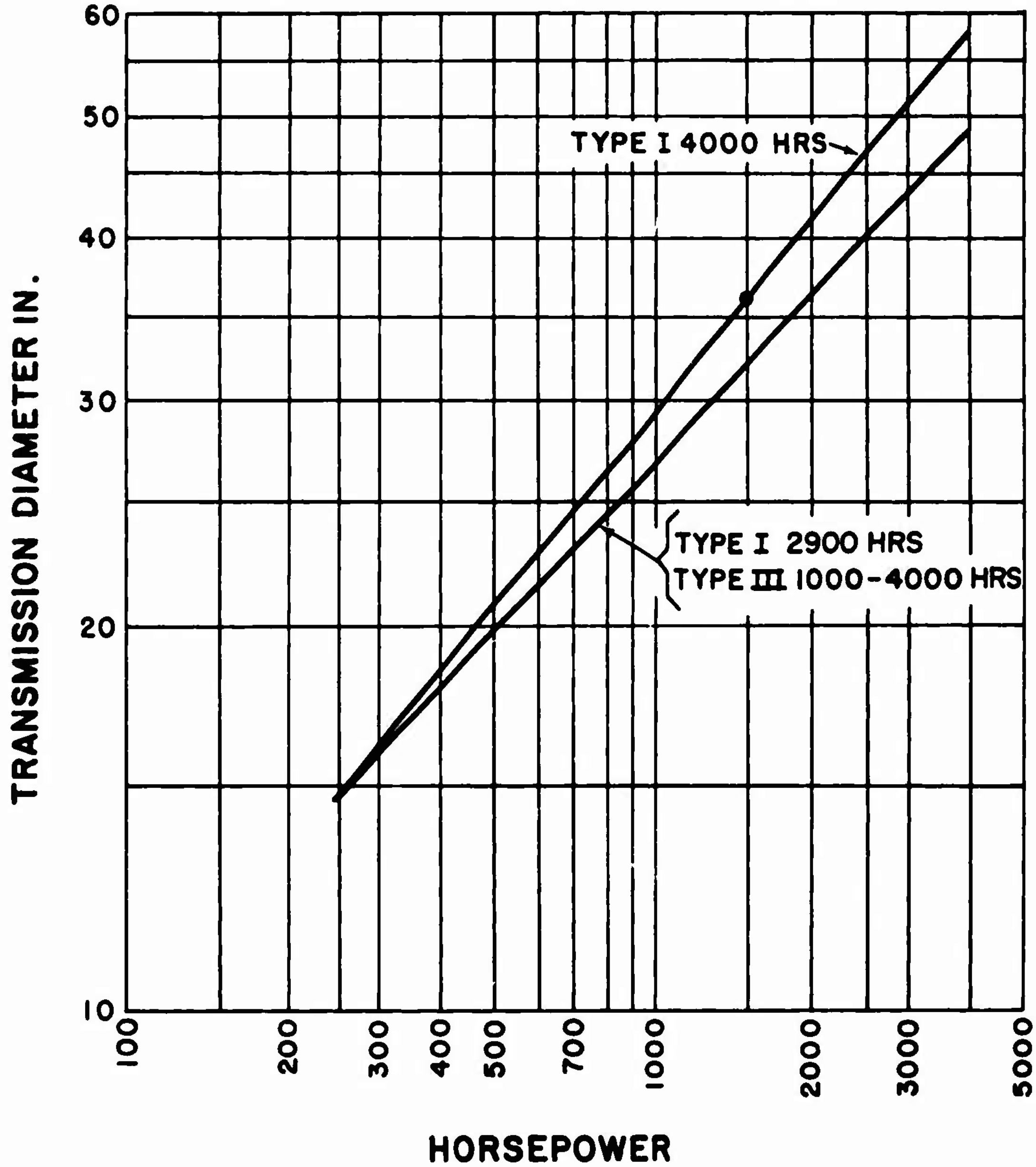


Figure 4. Transmission Diameter Vs. Horsepower

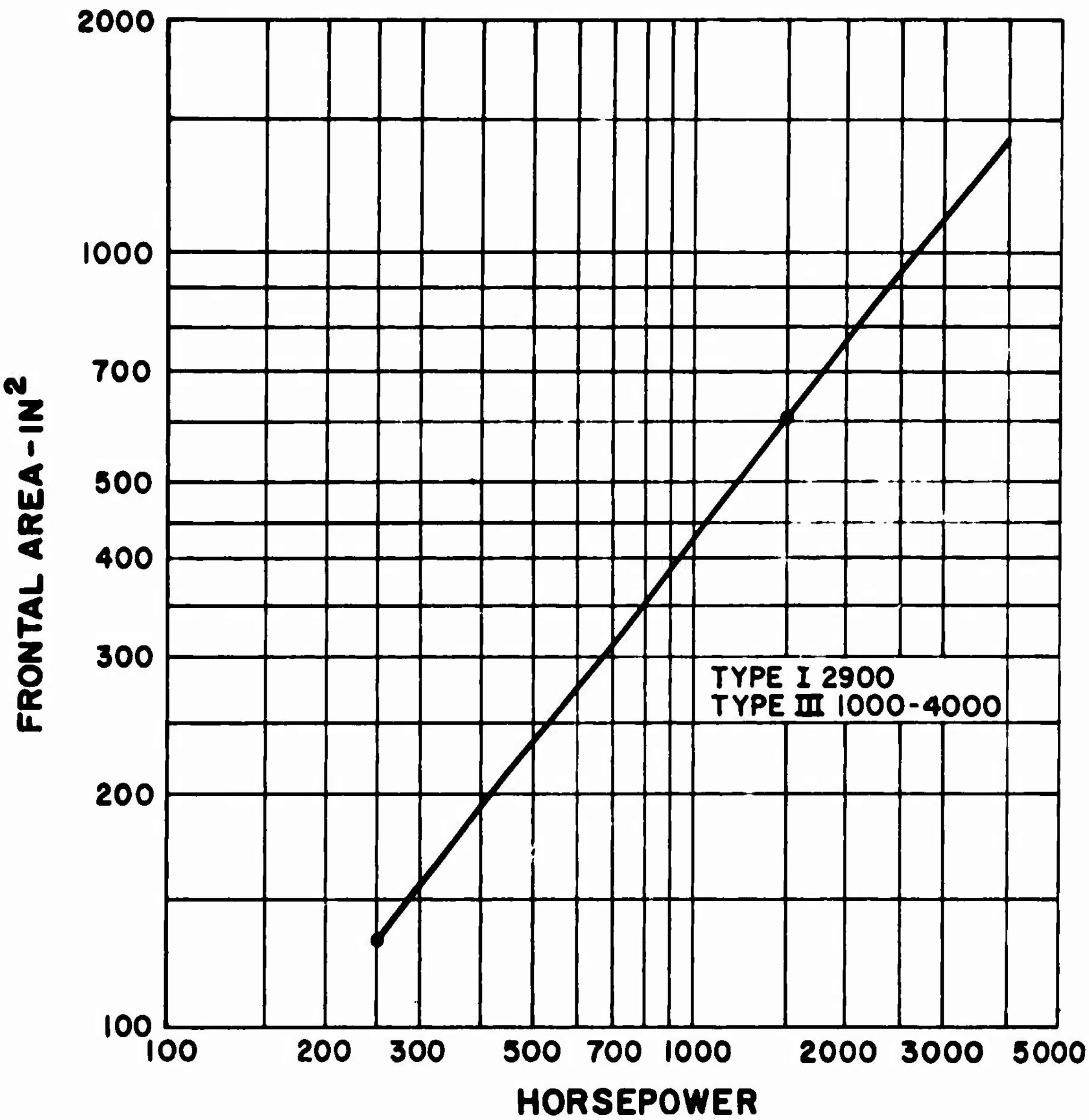


Figure 5. Frontal Area Vs. Horsepower

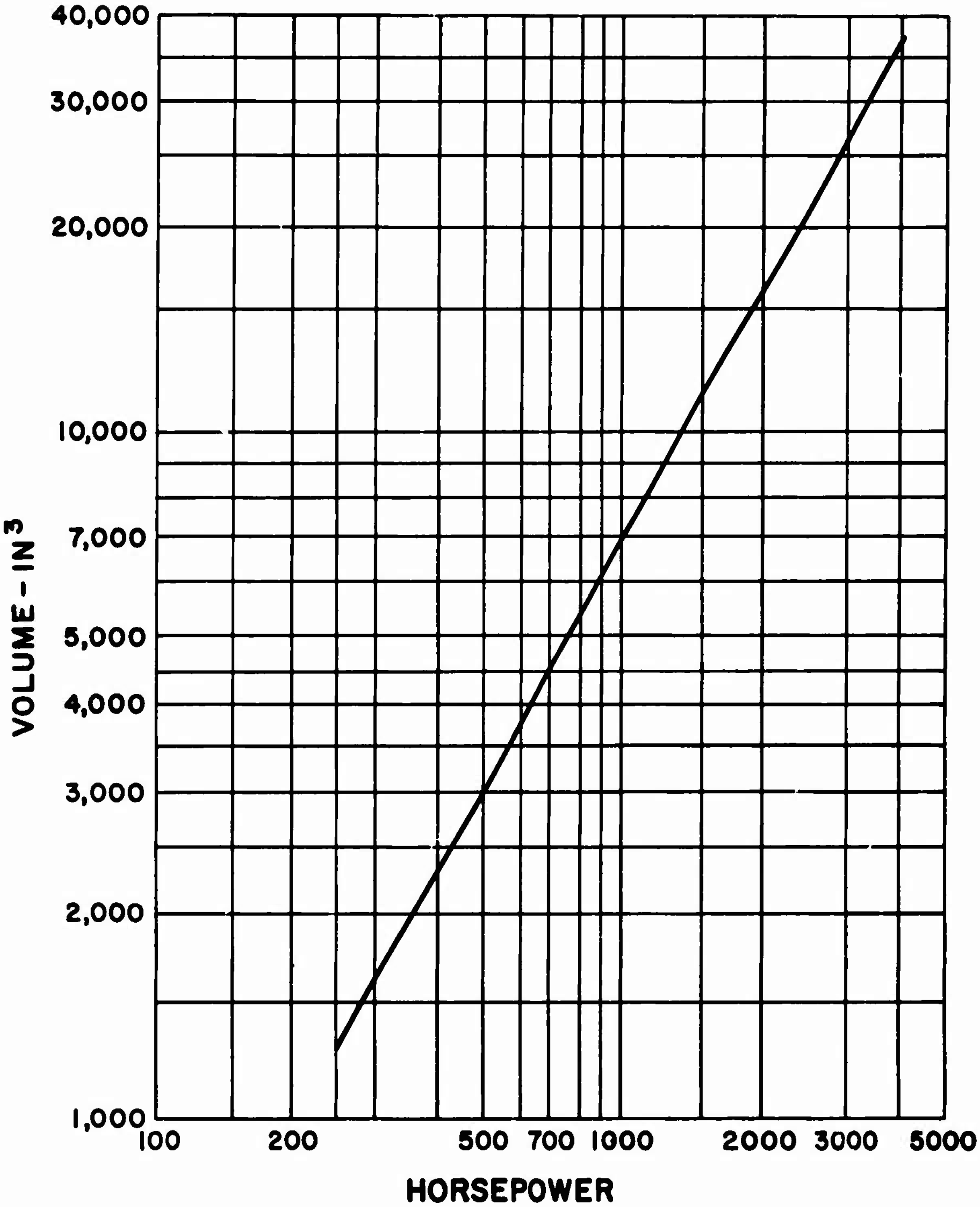


Figure 6. Transmissi_n Volume Vs. Horsepower

planet gear bearings must be accommodated under the secondary pinion rims, surrounding the bearing needles and for additional life it is necessary to increase the gear train size to accommodate larger bearings. Type I with the six trunnion mounted pinions is an efficient weight design since the trunnions and pinion rims automatically double as the inner and outer planet bearing races. The carrier configuration is conventional and readily provides sufficient rigidity to the pinion mountings to maintain proper gear alignment under torque. This design apparently offers the best opportunity for absolute minimum weight since so much of the planet gear and bearing weight performs dual functions. For helicopter application where the pinion RPM is low enough to permit needle roller bearings, it is particularly attractive. The diameter of the secondary pinion must be sufficient to accommodate adequate capacity needle bearings under its rims. Furthermore, as the inner race of the bearing must be sufficiently stiff, as a part of the carrier, to limit twist under load to an acceptable limit. This combination of characteristics is obtainable with well proportioned design as is proven in the results of the Design Variation Evaluation, Page 11 and the Parametric Design Evaluation, Page 33. In order to obtain 4000-hour bearing life capacity, it is found necessary to increase the overall diameter of the design in order to accommodate larger planet bearings under the rim of the pinions. However, the design capacity increases as the tenth power of the transmission diameter which indicates that it is not particularly difficult to accomplish.

The trunnion type carrier can readily straddle the gear train to provide an outboard bearing support for the output shaft. This is a weight and compactness advantage particularly attractive if the moments on the output shaft are minimal. As a matter of good design practice, all types considered support the rotor shaft independently so that the gear trains cannot be affected by rotor shaft moments.

Type III uses a completely tested and proven load equalization means. The input quill shaft splits torque into three paths at the primary mesh followed by a split into six loads at the opposed helical mesh in the secondary. The arrangements of the gears and bearings is such as to optimize carrier design since driving loads are symmetrical and thereby, impose no twisting moment on the carrier or pinion axes which would

tend to skew the mesh. The bearing dimensions are not limited by the rim under the secondary pinion teeth so there is more design flexibility in providing planet bearing capacity. Since there are only six bearings, there is an advantage to system life compared to a system that uses six pinion centers and twelve bearings. Due to the practical elimination of significant moments, a lightweight simple configuration is used.

In summation, for the purpose of building a better helicopter transmission that is suitable for ready integration into Army aviation, Type III is recommended. It is lightweight, uses the minimum number of dynamic parts, and provides automatic equalization by rolling dynamic gear mesh action.

DESCRIPTION OF CONCEPTS - FIGURE 1

The concept common to all three types is the principle of assuring load equalization in spite of tolerance variations in multiple power path compound planet gear systems. Normally, gear trains have spring rates with orders of magnitude in the hundreds of thousands of inch pounds torque per inch of accommodation. Since gear teeth deflect from zero to full load in approximately 0.001 inch equal load distribution is almost impossible with high rate systems. The concepts studied here are unique in that they provide spring rates reduced to approximately one thousandth of conventional values thereby, making possible significantly improved load path tolerance accommodation.

Types I and II accomplish this by equally dividing the input torque applied to six primary planet gear meshes. With the gear proportions required to provide large ratios the primary planet gears are so large as to preclude more than three in one plane so two sets of three primary planet gears are provided which mesh with a dual input sun gear. The dual input sun gears are mounted on the end of an extremely low radial spring rate quill shaft. These sun gears are mounted on three rocking levers which permit each sun gear to adjust torsionally relative to the other so that their total torques are equal. Furthermore, the rocking levers permit radial position accommodation so that its three planet gear loads will balance.

This system for dividing and balancing loads through rocking levers is called a "whiffletree" arrangement in deference to its time honored ancestor used for multiple hitching of horses. Naturally, as long as the load readily divides equally and accommodates tolerance differences at the primary mesh, it will also provide proper load division at the secondary reaction mesh.

Type I is the conventional design with a trunnion carrier mounting of the planet pinions wherein the twelve planet bearings are accommodated under the rim of the planet pinion. It has six primary and secondary gear meshes all spur.

Type II utilizes the whiffletree sun gear arrangement and six primary planet spur gear mesh points. The secondary mesh is further equally divided into twelve mesh points with dual opposed helical internal ring gears. The twelve compound planet pinion bearings are axially free so that the secondary mesh floats to the position of equalization unhindered by the primary spur mesh. The carrier and planet bearings are arranged so that twisting moment effects are cancelled or neutralized to provide optimum gear and bearing operating conditions.

Type III is a compromise between the other two types. It utilizes three compound planets meshing with a single quill mounted spur sun gear. The radial spring rate of the sun gear is extremely low so that it moves radially to divide the torque into three almost identical primary gear forces. These planet pinions, mounted in bearings which provide no axial positioning, mesh also with opposed helical ring gears (secondary). Responding automatically to nonuniform loading each of three pinions floats axially to the position where its torque load divides equally at the secondary mesh with the dual ring gears. The six planet pinion bearings are accommodated in a carrier configuration wherein the twisting moments are equalized and also the elastic flexibilities compensate so that the inside diameter to the outside diameter parallelism is retained throughout the torque range. This three-primary path and six-secondary load path system is the Type III configuration chosen for specific design in contractual Phase II.

DESIGN VARIATION EVALUATION

By use of the governing mathematical relationships and analysis of ratio requirements as discussed in Appendices I, II, III, and IV, design variations for Types I, II, and III have been investigated throughout the expected operating range. Basic transmission sizes have been determined for 250, 500, 1500, 2500 and 4000 horsepower at 17:1 ratio in the main transmission set. These values are tabulated in Table I. Preliminary layouts of the planetary gear and bearing arrangements for these conditions provide an accurate factor for overall size variation. The 250-horsepower and 1500-horsepower sizes in the three types are shown on Figures 7, 8, and 9.

Weight analysis of these designs including only the main dynamic elements reveals that Type I and III are lighter than Type II at the same life ratings. Therefore, further design evaluation is completed only on Types I and III. Weight data as completed to provide comparisons at two different powers and different life factors are tabulated in Table II.

The overall transmission weight factors as shown in the graphs of the Parametric Study Results, Page 2, are truly lightweight compared against other generally published information and specific studies conducted by this contractor. In particular, the data indicate that the design is inherently efficient enough weight-wise to permit actual consideration for flight installation use of 4000-hour system life expectancy designs.

This parametric study is concerned with the weight of the main reduction units required for a given application which means that the weight of tail rotor, accessory, and angle input gear sets is not a proper concern at this time. For the total transmission weight as tabulated previously and shown on the graphs of the Parametric Study Results, Page 2, the following rules govern.

1. The weight of the main reduction gear with its housings input shaft and output shaft (to the seal face) is complete as it could be for a flight installation.

2. Approximately 5 percent of the main transmission weight is required in the input and accessory section to accommodate the ratio reduction necessary to provide the overall ratio required.

			I Input Divides 6 Ways Thru the Whiffle-6 Output Take Offs	II Input Divides 6 Ways Thru the Whiffle-12 Output Take Offs	III Input Divides 3 Ways Thru Single Sun Gear-6 Output Take Offs
1	Output Speed	250	410		
		500	310		
		1500	205	Same as I	Same as I
	n = (rpm)	2500	170		
		4000	140		
2	Output Torque	250	38,400		
		500	101,600		
		1500	461,000	Same as I	Same as I
	t - (in-lb)	2500	926,500		
		4000	1,800,000		
3	Input Speed	250	6970		
		500	5270		
		1500	3485	Same as I	Same as I
	n = (rpm)	2500	2890		
		4000	2380		
4	Input Torque	250	2260		
		500	5980		
		1500	27,100	Same as I	Same as I
	t = (in-lb)	2500	54,500		
		4000	105,900		
5	A Shaft Speed	250	2099		
		500	1587		
		1500	1049	Same as I	Same as I
	n = (rpm)	2500	870		
		4000	716		
6	Brg Load	250	820		1790
		500	1570		3430
		1500	4350		9500
	F_{RA} (lb)	2500	7000		15,300
		4000	10,900		23,800
7	Brg. Load	250	1000		1690
		500	1915		3241
		1500	5300		8960
	F_{RB} (lb)	2500	8530		14,400
		4000	13,250		22,400
8	$\frac{F_{RA} + F_{RB}}{2}$	250	910	870	1740
		500	1742	1668	3335
		1500	4825	4615	9230
		2500	7765	7425	14,850
		4000	12,075	11,550	23,100
9	Max. Brg. Dia.	250	1.189	2.187	
		500	1.899	3.04	
		1500	3.489	4.92	Same as II
		2500	4.549	6.2	
		4000	5.849	7.72	
10	A Dia.	250	1.84		
		500	2.548		
		1500	4.140	Same as I	Same as I
		2500	5.207		
		4000	6.50		

B Dia.	250
	500
	1500
	2500
	4000
C Dim.	250
	500
	1500
	2500
	4000
D Dim.	250
	500
	1500
	2500
	4000
E Dim.	250
	500
	1500
	2500
	4000
Pinion Brg. F Dia. (o.d.) (Cat. Choice)	250
	500
	1500
	2500
	4000
Pinion Brg. G Dia. (i.d.) Cat. Choice	250
	500
	1500
	2500
	4000
H Brg. No. Cat.	250
	500
	1500
	2500
	4000
I Brg. Length	250
	500
	1500
	2500
	4000
Individual Brg. Component J Bio Life	250
	500
	1500
	2500
	4000
K D. P.	250
	500
	1500
	2500
	4000

A

TABLE I. BASIC TRANSMISSION SIZES

		I	II	III
B Dia.	250	5.740	Same as I	Same as I
	500	7.950		
	1500	12.916		
	2500	16.245		
	4000	20.280		
C Dim.	250	.55	1.02	1.02
	500	.761	1.412	1.412
	1500	1.237	2.295	2.295
	2500	1.556	2.886	2.886
	4000	1.941	3.600	3.600
D Dim.	250	1.200	1.94	1.94
	500	1.662	2.686	2.686
	1500	2.700	4.365	4.365
	2500	3.396	5.490	5.490
	4000	4.236	6.848	6.848
E Dim.	250	1.425	2.84	2.84
	500	1.973	3.93	3.93
	1500	3.206	6.39	6.39
	2500	4.032	8.03	8.03
	4000	5.030	10.02	10.02
Pinion Brg. F Dia. (o.d.) (Cat. Choice)	250	1.125	2.187	Same as II
	500	1.875	3.000	
	1500	3.750	4.375	
	2500	4.500	6.000	
	4000	5.750	7.500	
Pinion Brg. G Dia. (i.d.) Cat. Choice	250	.875	1.375	Same as II
	500	1.625	1.750	
	1500	3.000	3.250	
	2500	4.000	3.750	
	4000	5.000	5.000	
H Brg. No. Cat.	250	7174CR Orange	Torrington HJ-263516	Same as II
	500	7295CR Orange	HJ-364828	
	1500		HJ-607632	
	2500		HJ-729640	
	4000		HJ-9612048	
I Brg. Length	250	.750	1.00	Same as II
	500	1.000	1.75	
	1500	1.000	2	
	2500	1.250	2.5	
	4000	1.750	3.0	
Individual Brg. Component J Bio Life	250	1700	11,000	1100
	500	2900	30,000	3000
	1500	1875	12,500	1250
	2500	2900	17,000	1700
	4000	5000	13,800	1380
K D.P.	250	"	"	"
	500	"	"	"
	1500	"	"	"
	2500	"	"	"
	4000	"	"	"

		I
Planet Gear L Face Width	250	.400
	500	.554
	1500	.900
	2500	1.132
	4000	1.412
Sun Gear M Face Width WF	250	.34
	500	.494
	1500	.84
	2500	1.072
	4000	1.352
Output Mesh N Face Width 123T	250	.62
	500	.86
	1500	1.40
	2500	1.75
	4000	2.18
P Dim.	250	
	500	
	1500	
	2500	
	4000	
Q Dim.	250	
	500	
	1500	
	2500	
	4000	
R Dim.		.625
		.866
		1.41
		1.77
		2.21

Note

For application
see figures 7, 8

B

ANSMISSION SIZES

II	III
Same as I	Same as I
1.02 1.412 2.295 2.886 3.600	1.02 1.412 2.295 2.886 3.600
1.94 2.686 4.365 5.490 6.848	1.94 2.686 4.365 5.490 6.848
2.84 3.93 6.39 8.03 10.02	2.84 3.93 6.39 8.03 10.02
2.187 3.000 4.375 6.000 7.500	Same as II
1.375 1.750 3.250 3.750 5.000	Same as II
Torrington HJ-263516 HJ-364828 HJ-607632 HJ-729640 HJ-9612048	Same as II
1.00 1.75 2 2.5 3.0	Same as II
11,000 30,000 12,500 17,000 13,800	1100 3000 1250 1700 1380
" " " " "	" " " " "

		I	II	III
Planet Gear L Face Width	250 500 1500 2500 4000	.400 .554 .900 1.132 1.412	.400 .554 .900 1.132 1.412	.800 1.08 1.80 2.264 2.824
Sun Gear M Face Width WF	250 500 1500 2500 4000	.34 .494 .84 1.072 1.352	.34 .494 .84 1.072 1.352	.74 1.020 1.74 2.204 2.764
Output Mesh N Face Width 123T	250 500 1500 2500 4000	.62 .86 1.40 1.75 2.18	.31 .43 .70 .875 1.09	.62 .86 1.40 1.75 2.18
P Dim.	250 500 1500 2500 4000		4.56 6.315 10.260 12.90 16.09	4.94 6.84 11.11 13.98 17.43
Q Dim.	250 500 1500 2500 4000		5.28 7.312 11.88 14.94 18.63	
R Dim.		.625 .866 1.41 1.77 2.21	.77 1.06 1.76 2.22 2.78	

Note

For application of data shown see figures 7, 8, and 9

C

TABLE II. TRANSMISSION WEIGHT DAT

	TYPE I				
	250 H. P.		1500 H. P.		
Transmission Element	1000 Hrs. (1900 Actual)	4000 Hrs.	1000 Hrs. (2900 Actual)	4000 Hrs.	10
	Tot. Wgt.	Tot. Wgt.	Tot. Wgt.	Tot. Wgt.	T
Pinion Brg's	.60	.81	6.00	8.10	
Planet Gears	6.00	6.90	81.00	93.12	
Trunnion Hsg.	3.60	5.10	25.80	36.78	
Pinion Assy	3.00	3.53	49.80	58.70	
Carrier	14.40	15.40	94.11	100.70	
Ring Gear	3.30	3.40	40.80	42.40	
Sun Gear & Quill Shaft	1.50	1.55	20.40	21.60	
Middle Hsg.	3.00	3.85	28.60	37.05	
Output Shaft Brg. Support	3.00	3.15	35.10	37.00	
Upper Hsg.	5.00	5.30	55.50	58.25	
Output Shaft, Brg., Rear Pl't. Brg.	18.10	18.10	174.48	174.48	
Rear Hsg. Plate	2.50	2.85	18.45	21.15	
	64.00	69.94	630.04	689.33	
Weight Chargeable to Ratio Accommodation in Input Section	3.20	3.70	31.60	34.40	
Total	67.20	73.64	661.64	723.73	

A

TABLE II. TRANSMISSION WEIGHT DATA

TYPE I			TYPE III			
.	1500 H. P.		250 H. P.		1500 H. P.	
00 Hrs.	1000 Hrs. (2900 Actual)	4000 Hrs.	1000 Hrs.	4000 Hrs.	1000 Hrs.	4000 Hrs.
t. Wgt.	Tot. Wgt.	Tot. Wgt.	Tot. Wgt.	Tot. Wgt.	Tot. Wgt.	Tot. Wgt.
.81	6.00	8.10	.90	2.50	4.68	12.96
6.90	81.00	93.12	6.18	6.00	55.80	54.12
5.10	25.80	36.78	4.00	8.45	25.02	55.26
3.53	49.80	58.70	6.51	8.60	53.70	70.56
15.40	94.11	100.70	7.50	7.70	68.70	70.38
3.40	40.80	42.40	5.91	6.50	53.30	58.72
1.55	20.40	21.60	1.76	1.72	18.10	17.82
3.85	28.60	37.05	4.30	4.30	50.70	50.70
3.15	35.10	37.00	3.25	3.25	37.46	37.46
5.30	55.50	58.25	3.92	3.92	43.80	43.80
18.10	174.48	174.48	18.10	18.10	174.48	174.48
2.85	18.45	21.15	2.50	2.50	18.45	18.45
69.94	620.04	689.33	64.83	73.54	604.19	664.71
3.70	31.60	34.40	3.20	3.67	30.20	33.20
73.64	661.64	723.73	68.03	77.21	634.39	697.91

B

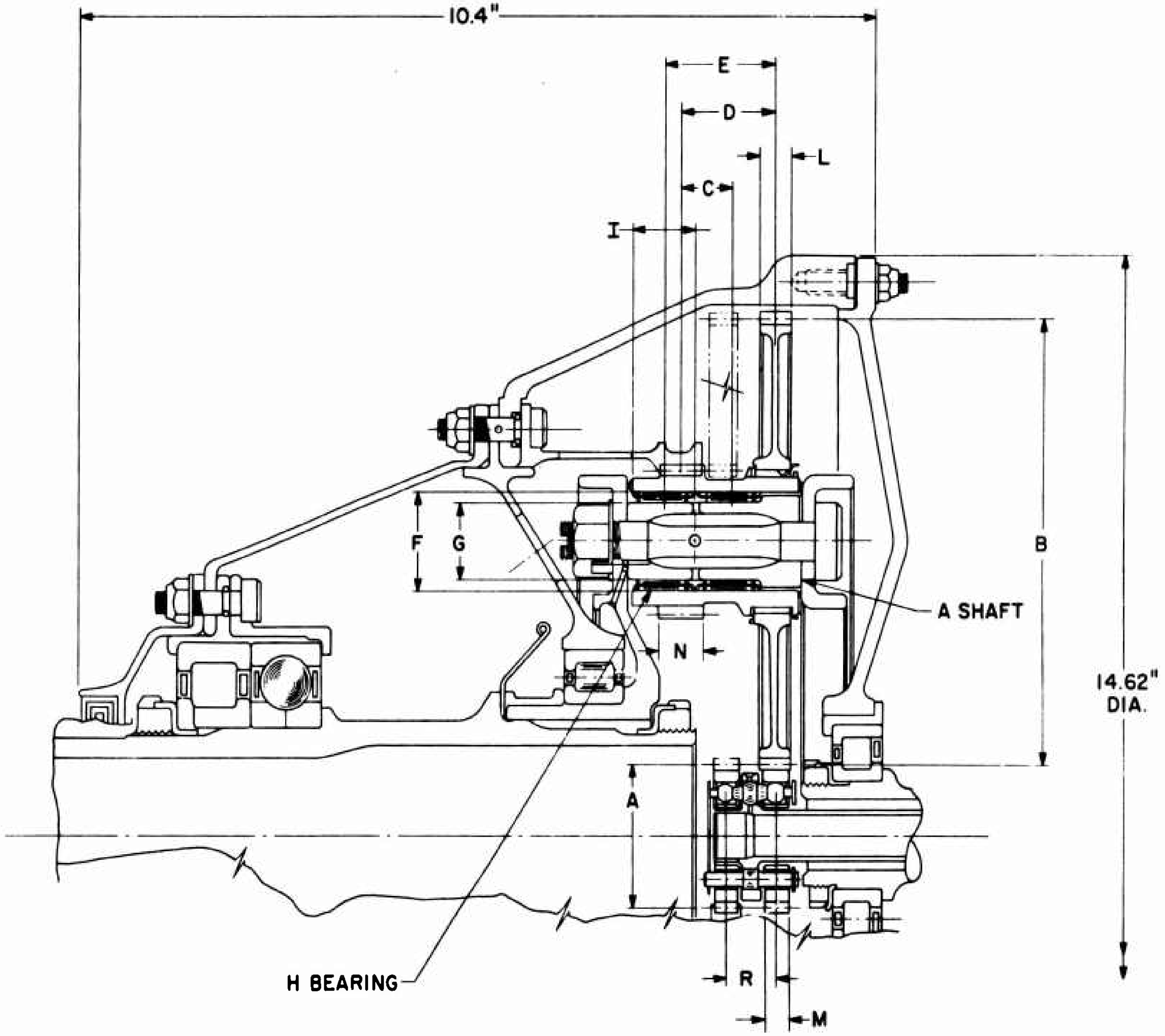
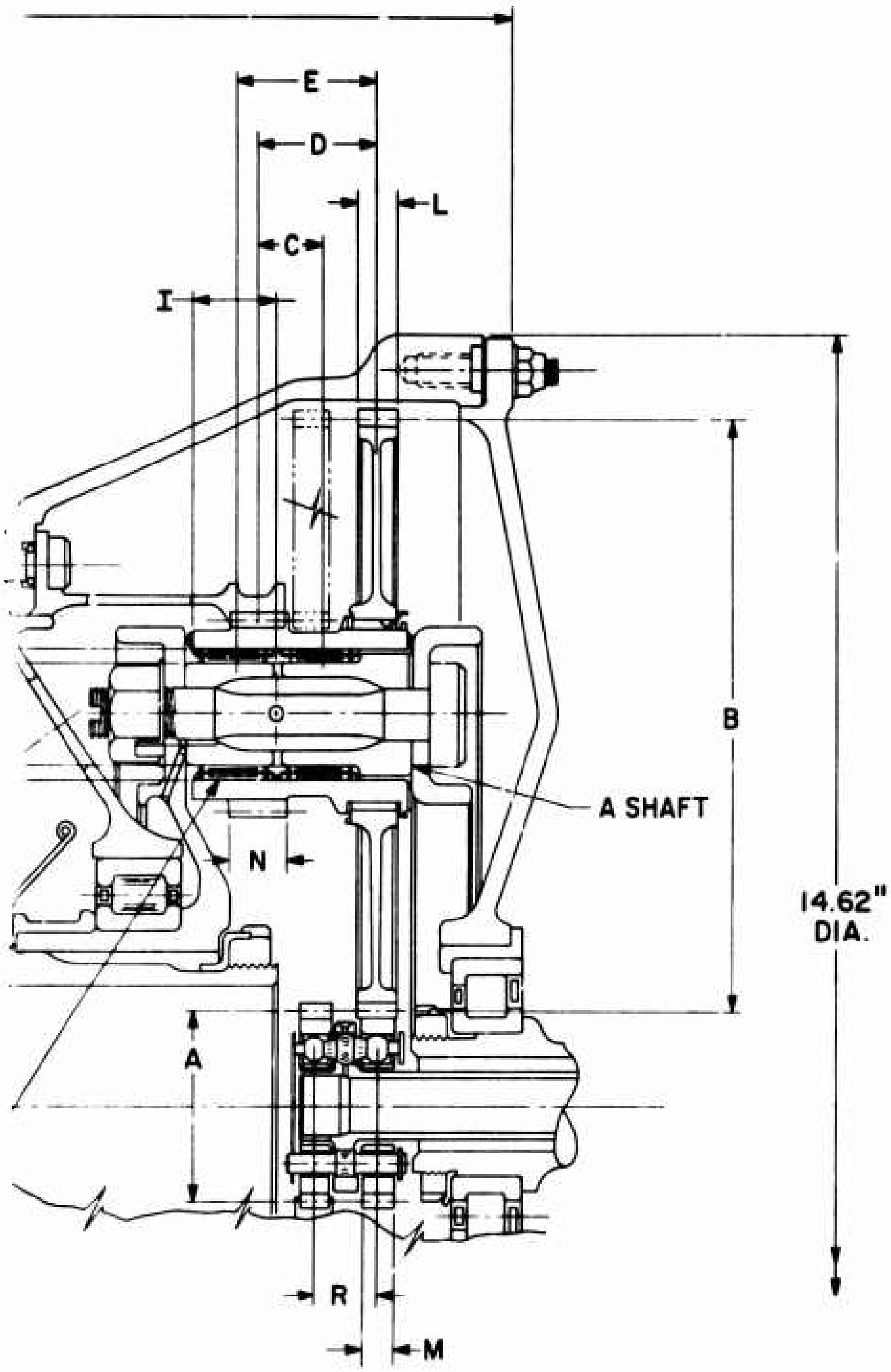


Figure 7. Type I - 250 Horsepower (1900 Hours)



TRANSMISSION WEIGHT TABULATION	
	TOT. WGT.
PINION BRGS	.60
PLANET GEAR	6.00
TRUNNION HSG.	3.60
PINION ASSY.	3.00
CARRIER	14.40
RING GEAR	3.30
SUN GEAR & QUILL SHAFT	1.50
MIDDLE HSG.	3.00
OUTPUT SHAFT BRG. SUPPORT	3.00
UPPER HSG.	5.00
OUTPUT SHAFT, BRG, REAR PLT, BRG.	18.10
REAR HSG. PLATE	2.50
	64.00
WEIGHT CHARGEABLE TO RATIO ACCOMMODATION IN INPUT SECTION	3.20
TOTAL	67.20

NOTE:
FOR MAGNITUDE OF DIMENSIONS INDICATED, SEE TABLE I

power (1900 Hours)

B

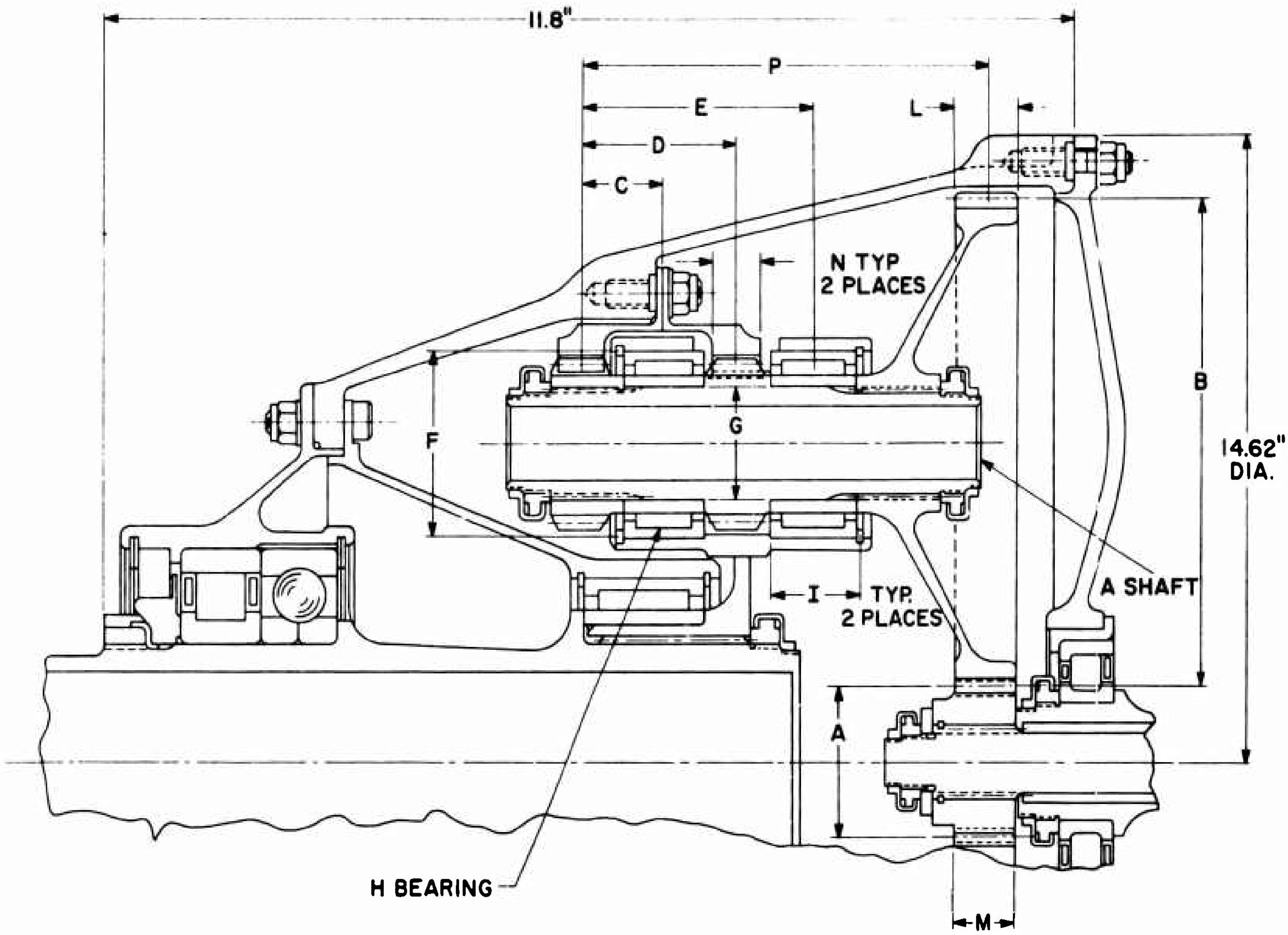
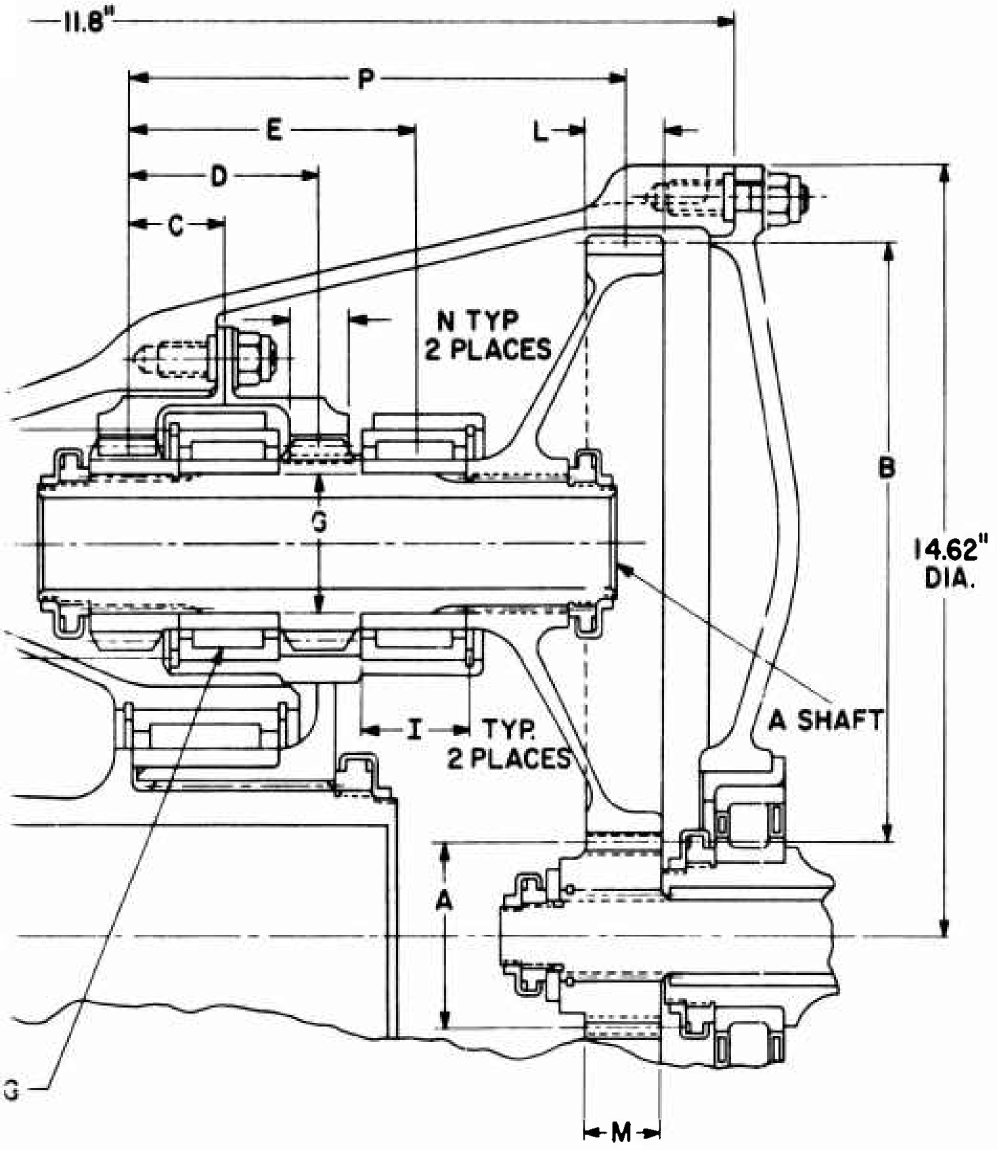


Figure 8. Type III - 250 Horsepower (1000 Hours)



TRANSMISSION WEIGHT TABULATION	
	TOT. WGT.
PINION BEARINGS	.90
PLANET GEARS	6.18
TRUNNION HSG.	4.00
PINION ASSY.	6.51
CARRIER	7.50
RING GEAR	5.91
SUN GEAR & QUILL SHAFT	1.76
MIDDLE HSG.	4.30
OUTPUT SHAFT BRG SUPPORT	3.25
UPPER HSG.	3.92
OUTPUT SHAFT BRG, REAR PLT. BRG.	18.10
REAR HSG. PLATE	2.50
	64.83
WEIGHT CHARGEABLE TO RATIO ACCOMMODATION IN INPUT SECTION	3.20
TOTAL	68.03

NOTE:
FOR MAGNITUDE OF DIMENSIONS INDICATED, SEE TABLE I

Horsepower (1000 Hours)

B

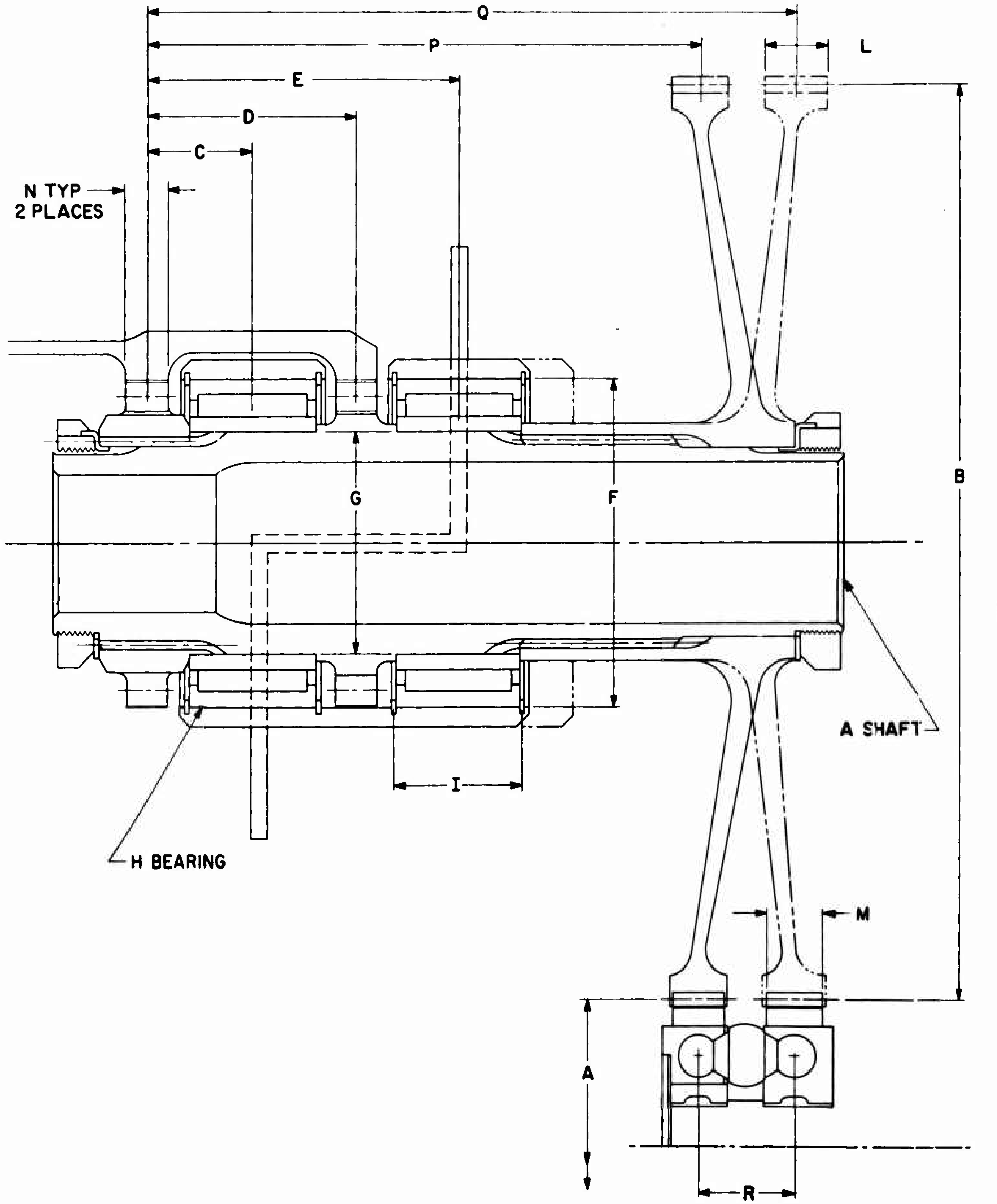
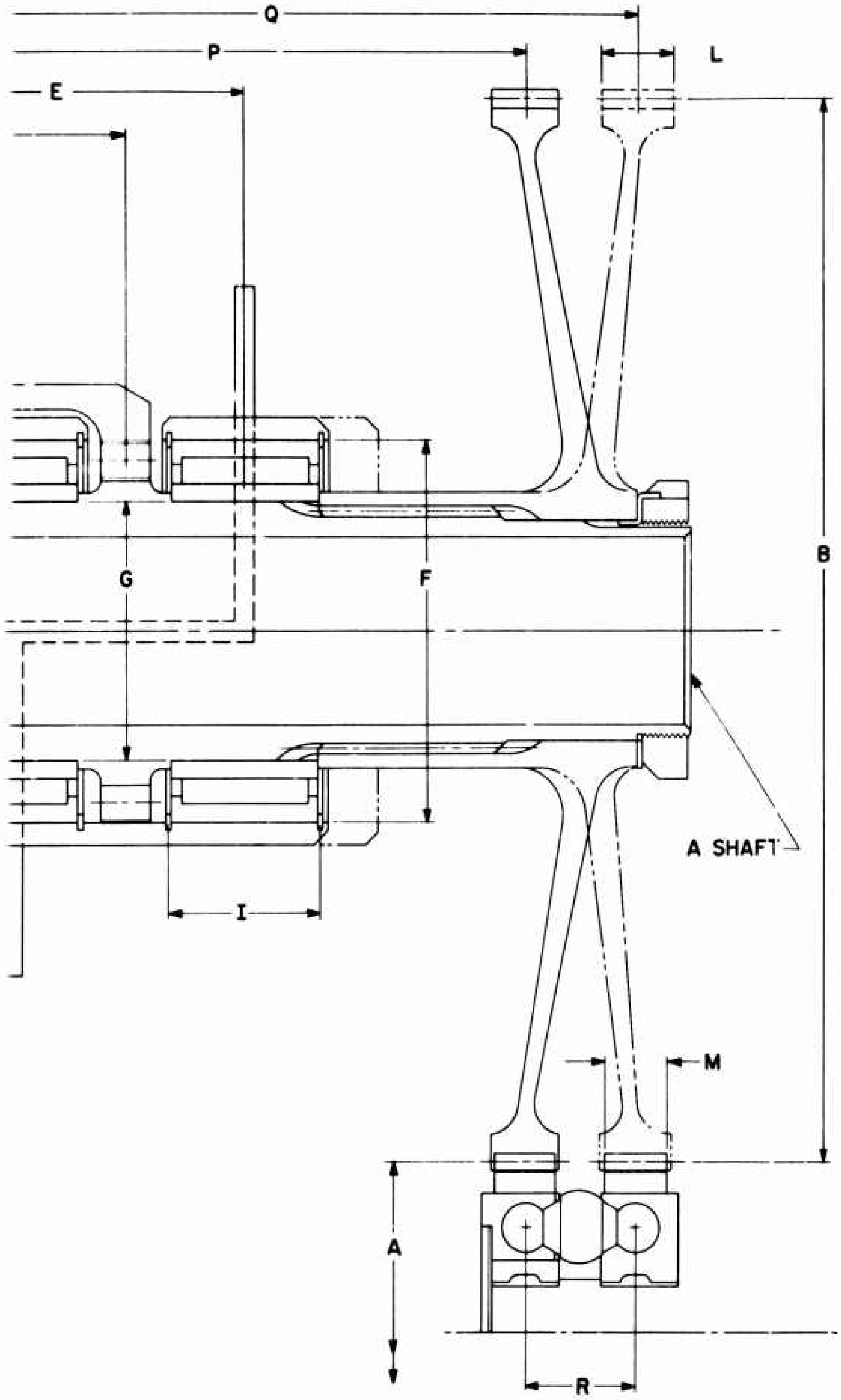


Figure 9. Type II - 1500 Horsepower (4000 Hours)



TRANSMISSION WEIGHT ANALYSIS	
	TOTAL WGT
PINION BEARINGS	12.24
PLANET GEAR	107.32
TRUNNION HOUSING	60.22
PINION ASSEMBLY	86.75
CARRIER	38.87
RING GEAR	36.56
SUN GEAR & QUILL SHAFT	20.18
TOTAL	362.14*

* ON COMPARABLE BASIS WEIGHT OF THESE PARTS:

TYPE I 296.73
TYPE II 362.14
TYPE III 299.57

NOTE:
FOR MAGNITUDE OF DIMENSIONS INDICATED, SEE TABLE I

'ype II - 1500 Horsepower (4000 Hours)

B

INPUT GEAR SET CONSIDERATION

The rules governing input gear sets are quite dependent upon the actual helicopter application. Since this parametric study is not concerned with actual flight applications, explanation is required regarding input gear sets.

1. It is mandatory to provide an angular input gear set to accommodate the horizontal turbine to the vertical transmission.

2. The main transmission must have attached to it an auxiliary section generally handling angular input gear set, tail rotor or interconnecting rotor drive and accessory drives.

With a main reduction optimized at say 25:1 or less overall ratio, as is the Bergen compound planet gear arrangement, many helicopter applications will require a ratio of from 1:1 to as much as 4.5:1 to be accommodated in the input gear set. Since an input set is required anyway, the only additional weight properly chargeable to the particular Bergen configuration is the weight to accommodate the ratio change as compared to 1:1 angular drive.

Input gear sets in the ratio of 1:1 to 4.5:1 can be either bevel, layshaft, planetary or star gear arrangements. From a weight efficiency standpoint the bevel sets are appreciably heavier than planetary arrangements. There is, as noted previously, however, no alternative to the use of angle input sets in helicopters.

The minimum weight design of the transmission should include 15:1 to 30:1 reduction in the planetary with the input set merely providing sufficient ratio to handle the supplied input speed.

With constant turbine input speed around 6000 RPM from 1:1 to 1.5:1 ratio input sets are required. With full turbine speed input approximately 4.5:1 ratio input sets satisfy all requirements.

Input bevel set designs available from previous studies substantiate the main conclusions regarding minimal weight penalty. Several layshaft input gear set arrangements in the 1500-horsepower and 250-horsepower range were figured to provide weight data, Figures 10 and 11.

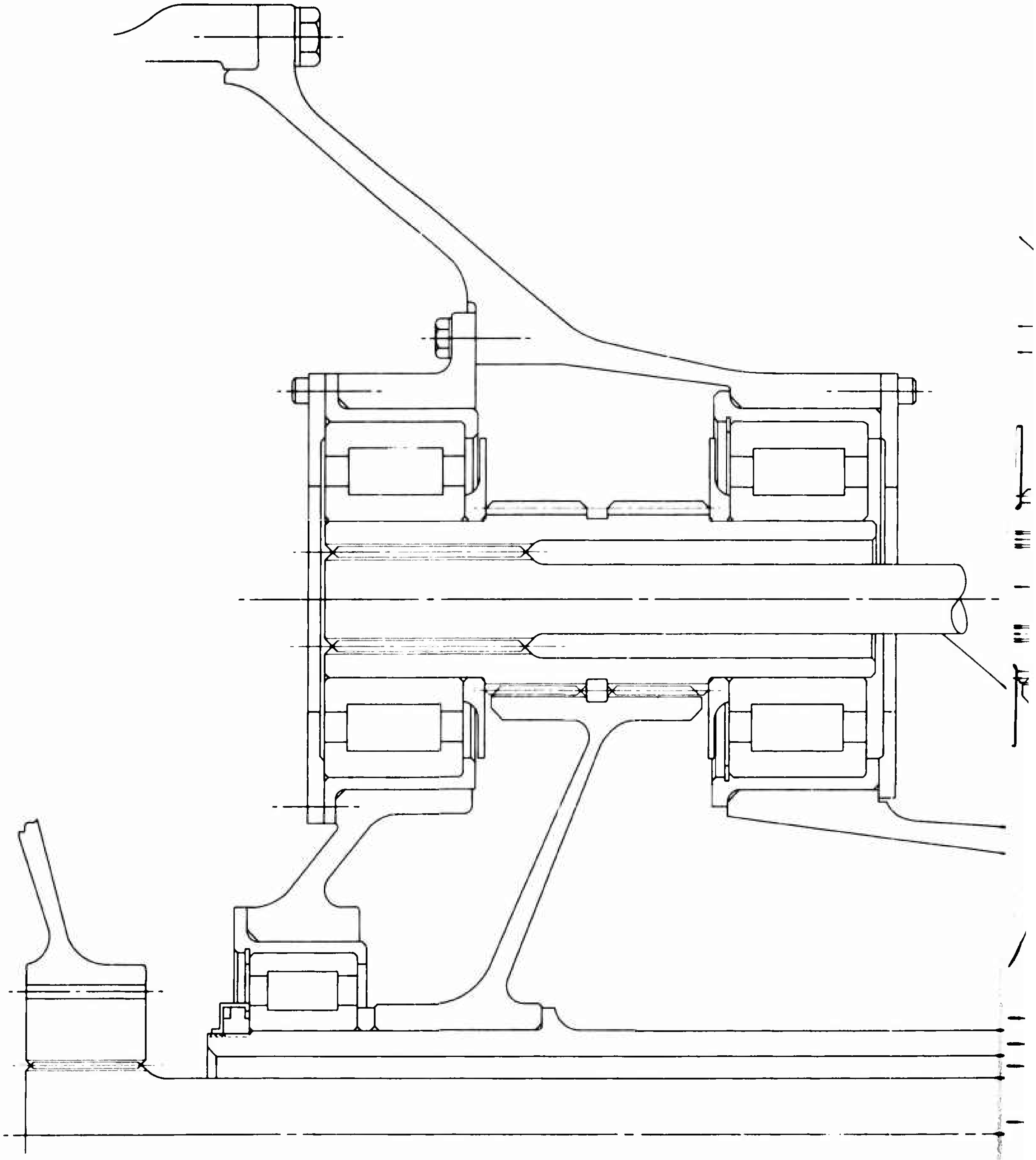
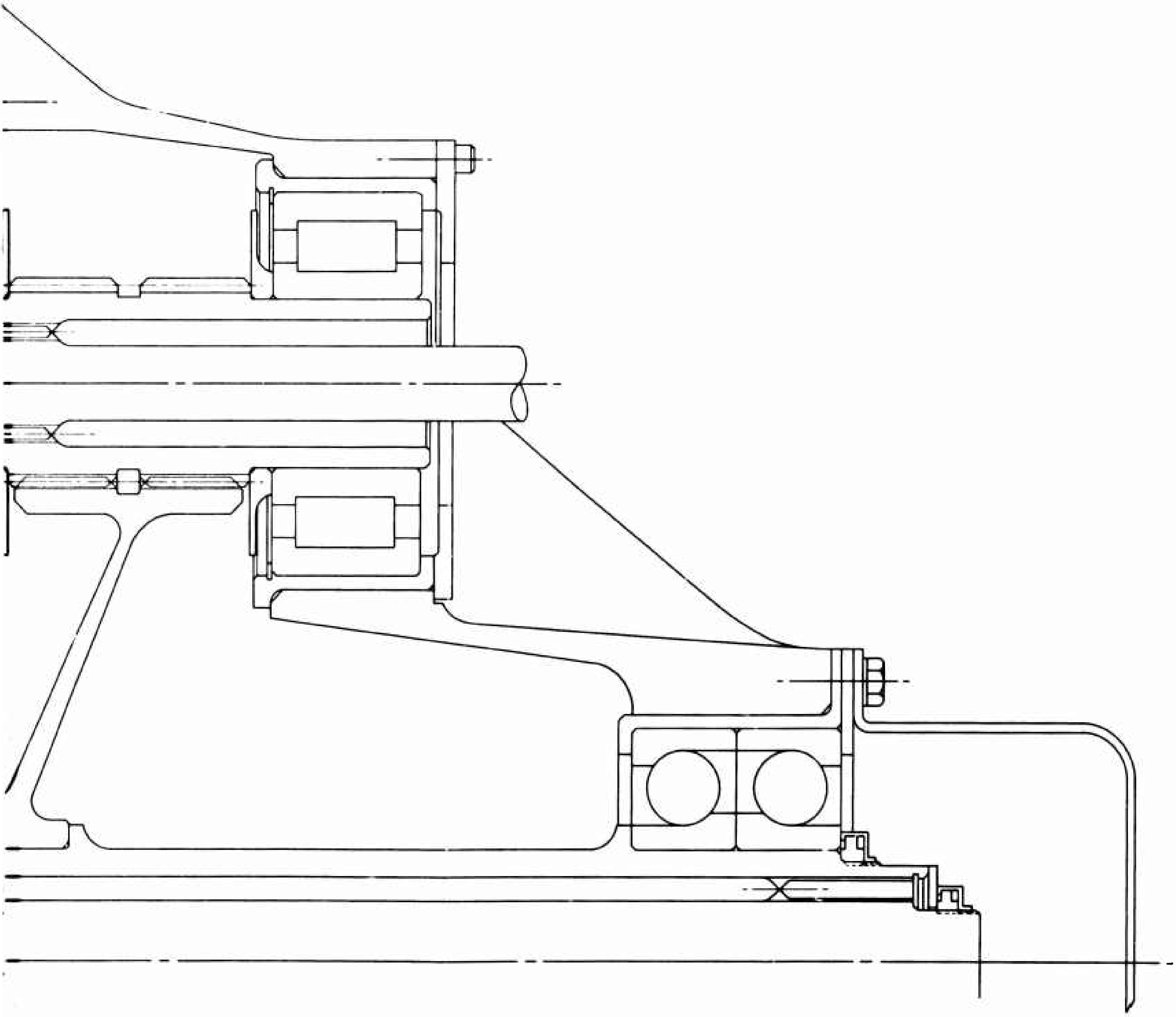


Figure 10. 5:1 Input Set Opposed Helical 1500 Horsepower

A



llcal 1500 Horsepower

B

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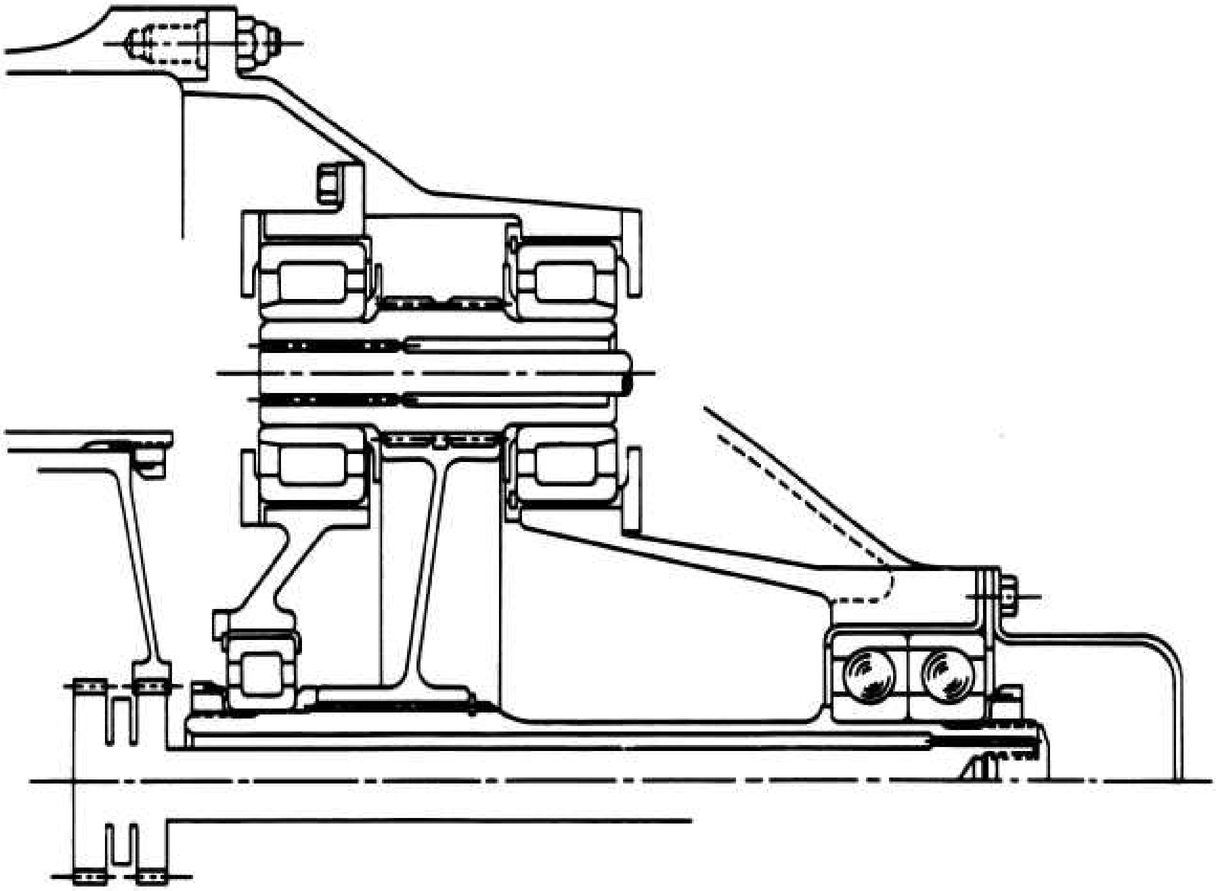


Figure 11. 5:1 Input Set Opposed Helical 250 Horsepower

250-HORSEPOWER DETAIL DESIGN

Upon approval of the Phase I study by the contracting officer in his letter dated 15 April 1964, the specific design of the three planet gear Type III configuration was initiated for Phase II. This was in accordance with the summary observation, Page 9, that the three-gear system appeared most advantageous for Army personnel at the contractor's facility on 9 April 1964, at which it was agreed that this Phase II design would have the following characteristics:

1. Rated power 250 horsepower (design power - 189 horsepower)
2. 17.1:1 ratio
3. 6,000 RPM input speed
4. Concentric input and output shafts
5. 4,000-hour - design life
6. 3-pinion compounded planetary gear train, Type III
7. Aircraft quality gears, bearing and shafts
8. Prototype test quality housing
9. Lubricant MIL-L-7808 or MIL-L-23699

This design has been completed and is shown in Figure 12. The gear train is set at a basic center distance of 3.835 inches with a 31-tooth sun gear mating with three 98-tooth primary planets; these are spur gears. The secondary gears are of double helical form and consist of three 22-tooth pinions meshing with a 112-tooth internal ring gear. The Hertz stress on both the primary and the secondary gears has been set at 160,000 PSI: This gives a design life on the primary of 6000-hours and an infinite life on the secondary gear set. The tooth fillet stress on both the sun gear and on the secondary pinion is 50,000 PSI which is below the endurance limit for the AMS-6265 material specified for these gears. The fillet stress on the sun gear and on the ring gear are lower. The scoring temperature rise on the primary gears is 58°F, which, with an oil-in temperature of 150°F, gives a scoring probability of about .03 percent (7).

Ref: (7) AGMA Tentative Information Sheet #217.01

The sun gear and the secondary pinion material is AMS-6265, the primary planet is made of AMS-6260 and the secondary ring gear material is AMS-6475. Conventional aircraft quality heat treatment is specified for these steels.

The primary planet is coupled to the secondary pinion by using a cropped section of the secondary tooth as a helical spline. Shims at this coupling are used to refine the angular alignment between the driving faces of the primary planet and the secondary pinion; this is done to assure equal driving load distribution without excessive sun gear radial movement or an excessively close alignment tolerance between the gear teeth and the spline teeth on the primary planet.

Journal extensions of the secondary pinion form the inner race of the planet bearings. The center of these bearings are spaced so that their load due to the gear driving force are equal; this assures equal design life for all planet bearings. Loaded deflection of the pinion shaft is 0.00018 inch/inch at the bearing centerline. This deflection angle is exactly balanced by the loaded deflection of the planet carrier members that mount the outer race of the planet bearings, thus the net deflection recognized by the bearing assembly is zero.

The planet carrier is a three-piece assembly, the two end sections are identical, each providing mounting for three planet bearings. The center section couples to the output shaft in such a manner that the total reaction load on each carrier and section is identical, thus, their deflections are identical. This further assures equal bearing life and prevents any twist of the carrier under load that might eccentrically load either the bearings or the gears; such an eccentric deflection would seriously reduce component life.

The planet bearings are precision caged needles that have been selected to have a system design life of 4,000 hours according to the method outlined in Appendix II.

This construction holds the planet assemblies rigidly in the radial direction; therefore, to assure equal driving load distribution to each of these three planets, both the stationary ring gear and the input sun gear must be radially free at a net spring rate that is insignificant compared to the gear separating force. The ring gear is mounted to

the housing through a loose fit double spline coupling that allows complete radial freedom within the limits required for load equalization. The sun gear is on the end of a splined quill that has a lateral rate of 2.5 pounds per 0.001 inch; this is insignificant compared with the 285-pound gear separating load and will have a 4-percent load distribution error at 0.010-inch gross pitch line spacing error.

Lubricant is pressure fed into the output shaft as indicated in Section D-D, Figure 12; the carrier is drilled to take this oil to the three carrier assembly bolts and thence to each of the six planet bearings. Jets are provided in the carrier center section to lubricate both the primary and secondary gear mesh. Separate lubricant input fitting is used to provide oil to the output shaft bearings. For a test setup, it is intended that oil will be supplied by an external motor driven pump.

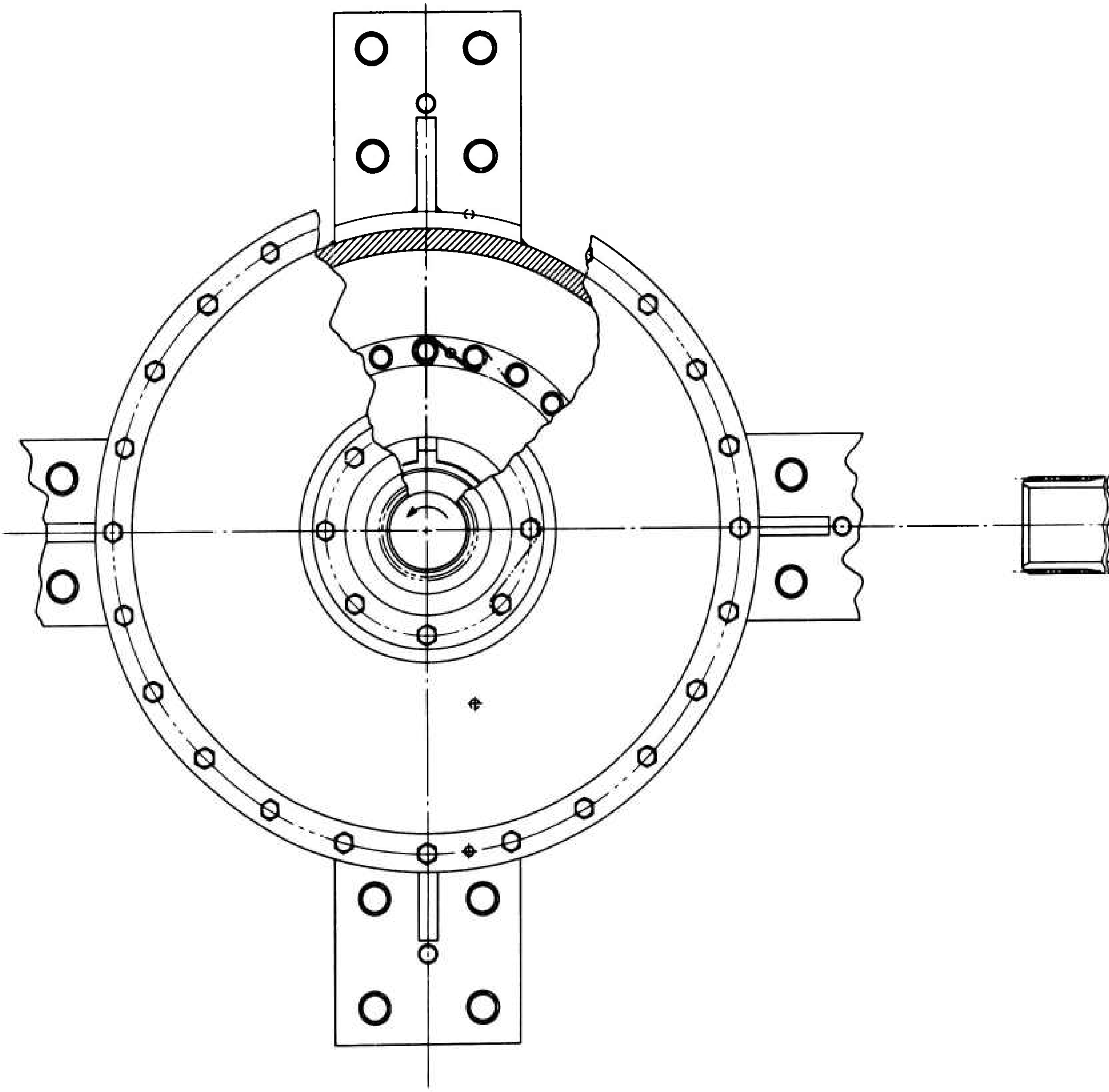
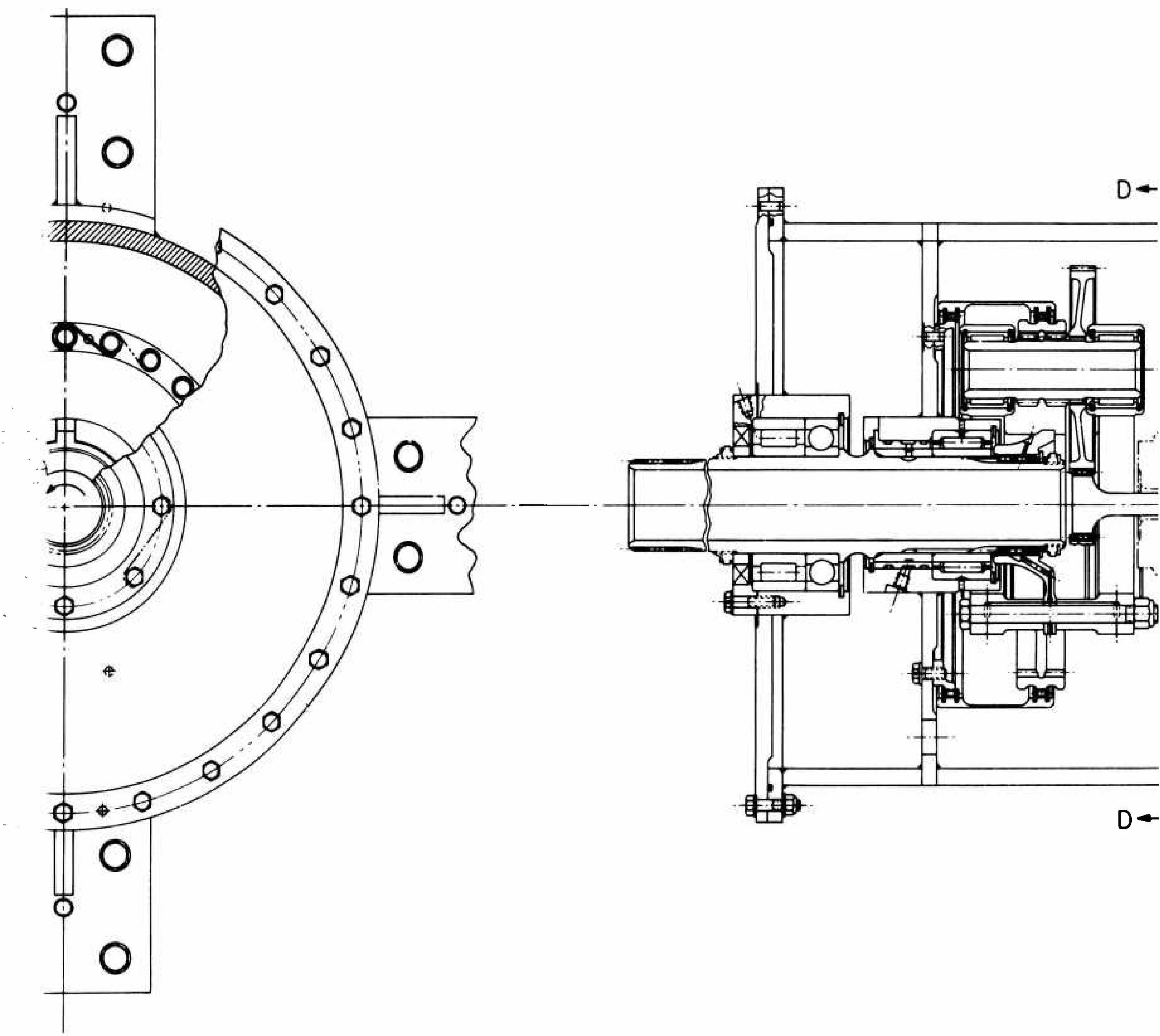


Figure 12. 250-Horsepower Detail Design

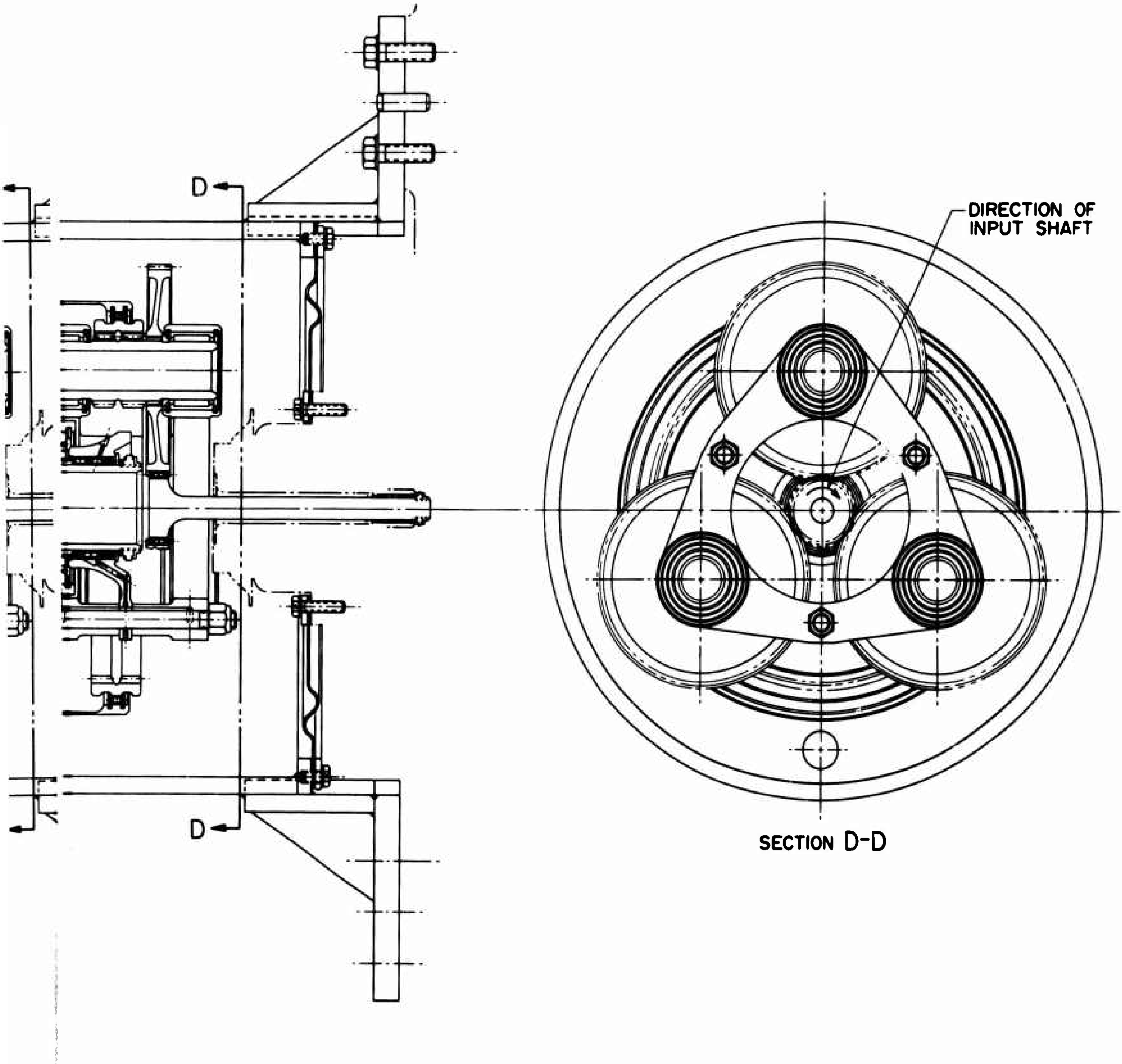
A



ɪrsepower Detail Design

31

B



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PARAMETRIC DESIGN EVALUATION

Certain basic design criteria for this type of compound planetary transmission may be set down in strict mathematical form together with constraints for the anticipated application.

This program has these inputs:

1. Rotor RPM
2. Design life - hours
3. Number of planet pinions
4. Rotor torque - inch-pounds RMC
5. Gear tooth hardness - BHN
6. Planetary speed ratio
7. Maximum transmission diameter - inches
8. Internal transmission temperature - °F.
9. Coefficient of gear tooth friction
10. Gear tooth surface finish - inches RMS

The program outputs are:

1. Diameter ratio of secondary pinion to sun gear
2. Sun gear diameter
3. Secondary pinion diameter
4. Primary face width
5. Secondary face width
6. Number of teeth in all gears
7. Scoring temperature on primary
8. Roll angle to scoring temperature
9. Scoring temperature on secondary
10. Roll angle to scoring temperature
11. Transmission merit factor

Internal constraints of the program are:

1. No gear shall have less than 18 teeth.
2. Tooth numbers shall be integers that will mesh on the specified number of trunnions.
3. There shall be enough diameter under the root of the sun gear and over the input shaft to place a whiffle assembly.
4. There shall be enough diameter under the root of the secondary pinion and enough length on the trunnion to place a roller bearing of the specified life.

The transmission gears designed by this program are considered to be life limited by the compressive stress level at the rolling pitch diameter (1) according to:

$$S_c = \frac{770\ H_b}{n^{.5}}$$

for the external gear mesh and

$$S_c = \frac{670\ H_b}{n^{.5}}$$

for the internal mesh where

H_b is the hardness of the gear teeth - BHN
n is the total number of cycles to initial pitting

The fillet tensile stress is calculated by the modified Lewis method using an empirically formulated J factor that assumes both a variable pressure angle and a variable tooth thickness; the limiting fillet stress level (2) is held to:

$$S_t = 1115\ H_b^{.6}$$

In certain cases, the tooth numbers determined by this stress will not mesh on the specified number of trunnions; tooth numbers are then reduced until proper meshing is obtained, with a corresponding reduction in fillet stress.

Gear tooth scoring characteristics have been included in this program by incorporating the Kelley flash temperature factor (3). This relation is:

$$t_f = \left[t_a + \frac{.0258f}{\text{Cos}\ \phi} \frac{W_t \left(\sqrt{V_1} - \sqrt{V_2} \right)}{F \sqrt{b/2}} \right] \left[\frac{50}{50-S} \right]$$

Ref: (1) Dudley "Practical Gear Design", Page 134
(2) Dudley "Gear Handbook", Pages 13-31
(3) Kelley "A New Look At The Scoring Of Gears" SAE Trans. 1963, Page 175

where

t_a	is the transmission internal ambient - ^oF.
f	is the coefficient of gear tooth friction.
W_t	is the gear tooth load - pounds
V_1 & V_2	is the gear profile rolling velocity - f.p.s.
ϕ_t	is the rolling pressure angle.
F	is the active face width - inches.
b	is the width of Hertzian contact - inches.
S	is the tooth surface finish - r.m.s. microinches

In the present study all gears are considered to be spur gears. Those that have a small helix angle to obtain longitudinal load balancing do not have enough helical overlap to negate this simpliciation.

Thus, the specific area of interest for scoring resistance is the area of two tooth contact where the tooth sliding velocity ($V_1 - V_2$) is greatest. Within this area it has been assumed that the tooth load (W_t) will buildup and decay linearly with a change in the gear roll angle, θ_a or θ_b. It would be more realistic to consider this load decay to vary as a function of change in gear roll angle that has a zero rate of change of load at the two end points, such as:

$$Z = 1 - 3X^2 + 2X^3$$

where

Z	is the ratio of load change
X	is the ratio of gear roll angle change

Although this type of relation would obtain more realistic values of t_f, the added complication to the program is not warranted at this time. This view is substantiated by inspection of the computed results for t_f, which all lie well below any possible critical limit (4); for the lubricant, MIL-L-7808, that is anticipated for these transmissions, this limit is about 330^{o}F.

The constraint for the minimum size for the input sun gear is determined by three stresses at the whiffle assembly:

1. The Hertz stress at the center section of the whiffle shall be no more than 19K p.s.i.

Ref: (4) Dudley "Practical Gear Design", Page 143

2. The bending stress of the whiffle shall be less than 76K p.s.i.

3. The shear stress in the fillet of the quill shall be less than 86K p.s.i.

Designs for this section have shown that these requirements can be satisfied if the pitch diameter of the sun gear is equal to or greater than:

$$D_1 = \frac{(Q_1)^{1/3}}{9 - 67/N_1}$$

where

Q_1	is the input torque - inch pounds
N_1	is the number of teeth in sun gear

The constraint for the minimum size of the secondary pinion D_4) presents a slightly more complicated situation. The shiffled six pinion setup can best be configured with the trunnion bearings within the bore of the secondary pinion, therefore, this bore and consequently the pitch diameter of the pinion must be large enough to accept bearings of the specified life. Also, the trunnion must have sufficient length for the bearing roller assembly.

The basic relation for roller bearing capacity is (5):

$$C = f_c \, (i \; l_b)^{7/9} \, Z^{3/4} \, d_r^{29/27}$$

where

f_c	is the geometry factor
i	is the number of rows per bearing
l_b	is the length of one roller - inches
Z	is the number of rollers per row
d_r	is the roller diameter - inches

The geometry factor (f_c) may be optimized at a value of 6550 if the bore of the pinion is made:

$$D_b = 6.55 \, d_r$$

Ref: (5) AFBMA Std. II, Page 1

In this case:

$$Z = 12$$

for the proper roller spacing to provide for a well proportioned roller retainer.

Overall bearing length is assumed to be satisfied by:

$$l_b = \frac{2F_1 + F_4}{2}$$

where

F_1 is the primary face width - inches.
F_4 is the secondary face width - inches.

Combining these conditions, the minimum pitch diameter of the secondary pinion is:

$$D_4 = \frac{\left[L\left(\omega_4 + \omega_1/m_g\right)\right]^{.28}\left(W_{t1} + W_{t4}\right)^{.93}}{4.66\left(2F_1 + F_4\right)^{.725}\left(1-{}^{8}/_{N_4}\right) \times 10^4}$$

where

L is the design life - hours
ω_4 is the secondary pinion speed - r.p.m.
ω_1 is the sun gear speed - r.p.m.
m_g is the transmission ratio
W_{t1} is the primary tooth load per pinion - pounds
W_{t4} is the secondary tooth load per pinion - pounds
N_4 is the number of teeth in secondary pinion

A transmission design merit factor, based on weight, has been included in this program. This merit factor (M_t) is derived from pre-existing design weight data on this type of transmission; as these data are quite incomplete at this time, this merit factor is, of necessity, quite coarse. Further refinement will surely accrue when more extensive and accurate input data become available; however, for present purposes, this merit factor will show certain gross aspects of design weight trends for this configuration to indicate means for minimizing this merit factor to obtain the lightest design.

The gross design weight of this transmission is attributable to three subsystems:

1. The primary gear set
2. The secondary gear set
3. The planet bearings

thus

$$M_t = K_1 M_1 + K_2 M_2 + K_b M_b$$

where

M_1 is the weight factor for primary gear set
M_2 is the weight factor for secondary gear set
M_b is the weight factor for the planet bearings
$K_{1,2,b}$ is the approximate constants

Given sufficient data on existing designs, the constants may be evaluated to obtain the generalized form of the merit factor for this type of design; this, in turn, may then be utilized to estimate the merit of similar designs. This has been done; the merit factor that is utilized is:

$$M_t = .075\, F_1\, D_1^2 \left(1 + G m_{g1}^2\right) + F_4\, D_4^2 \left(\frac{G - m_{g2}^2}{D_4}\right) + .016 G M_b$$

where

$$M_b = \left[L(\omega_0 + \omega_4)\right]^{.45} \left[W_{t1} + W_{t4}\right]^{1.5} \times 10^{-6}$$

G is the number of trunnions
m_{g1} is the primary gear set ratio
m_{g2} is the secondary gear set ratio
D_4 is the secondary pinion pitch diameter - inches
ω_0 is the output speed - r.p.m.

Discretion should be used in the interpretation of the resulting values of (M_t) obtained from this relation as no claim is made for absolute accuracy; it does, however, give relative values for gross trends.

Aside from the actual computed output data listed in the first part of this section, some data reduction has been made to determine significant trends for the merit factor as a function of other parameters. These are shown in Figures 13, 14, 15, and 16. The first of these shows that the trend of merit factor is nearly linear with output torque; this agrees well with other published data. The second figure indicates a variation as the 1/6 power of a change of design life. The comparable variation with a change in transmission diameter is to the 1/5 power. The last figure shows that the merit factor will vary as the 2.3 root of the size ratio of secondary pinion to sun gear. These relations may be summarized by the indication that a lighter transmission may be obtained by:

1. Lower output torque
2. Lower design life
3. Smaller diameter
4. Higher secondary ratio

Notable by its absence from the above discussion is any factual data on the effect of ratio change. Data that were obtained are obviously invalid and will require further refinement of the manner of obtaining the merit factor before they will be useful.

Some other interesting points can be brought out concerning this program. It should be noted that the compressive stress-life relation for internal gears differs from that for external gears. This was done not on the basis of gear life but rather in the interest of obtaining high enough tooth numbers in the secondary pinion. If the higher stress level had been used, many sets would have been rejected by the minimum tooth number constraint; this factor is peculiar to this configuration.

Consistency of these designs is further shown by the narrow spread in values for the scoring temperature, from 196°F to 223°F in the primary and from 192°F to 201°F in the secondary. In fact some of this spread can be attributed to the reduction in tooth numbers required for meshing; this reduction will raise temperature.

When work was initiated on this program, a search was made of some computer program libraries and conversations were had with many authorities in the field of gearing and computing. Nowhere could a similar program be found, nor could much background data be obtained that would establish a precedent. Therefore, this single effort must be

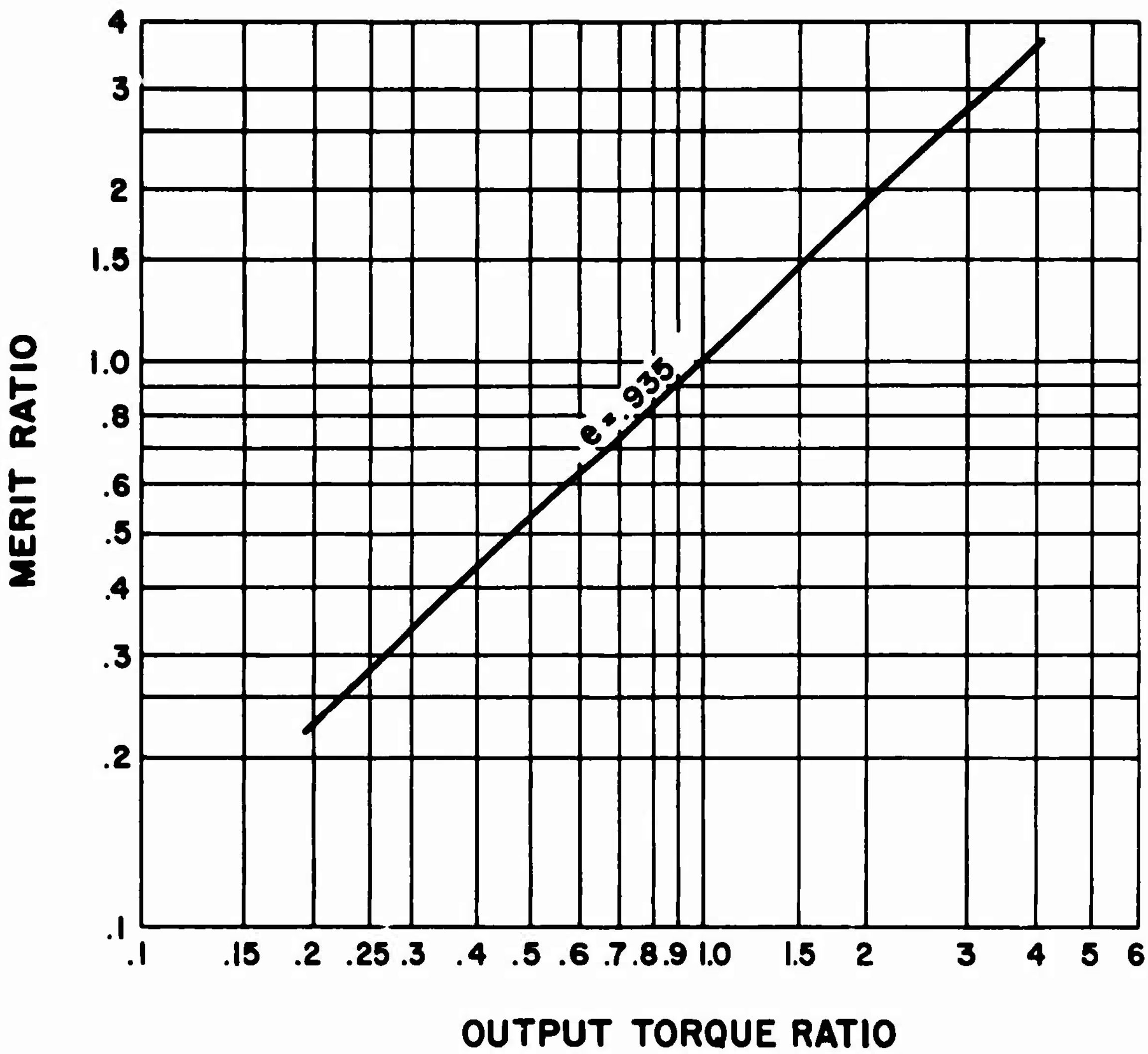

Figure 13. Merit Factor as Function of Output Torque Ratio

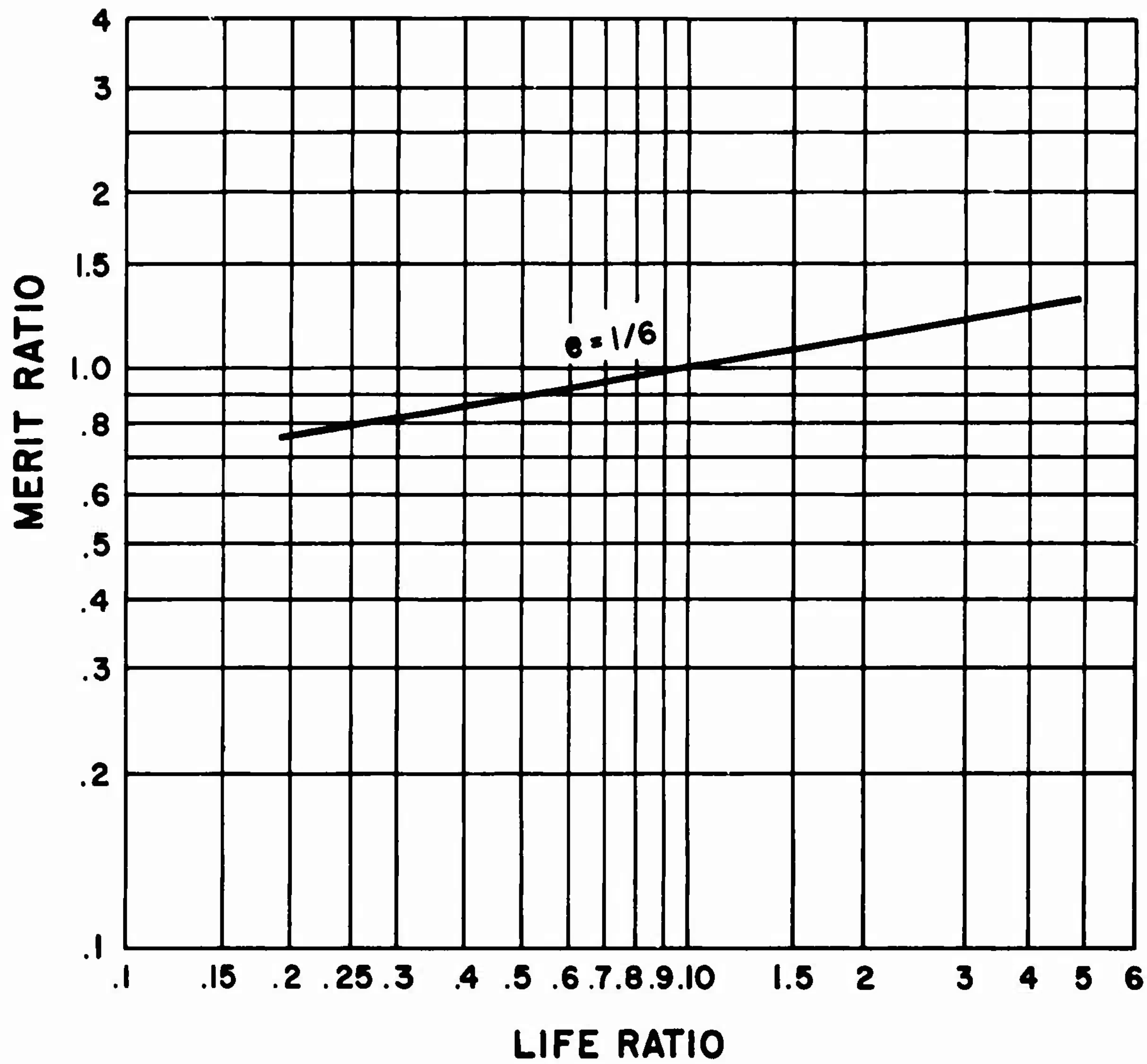

Figure 14. Merit Factor as Function of Life Ratio

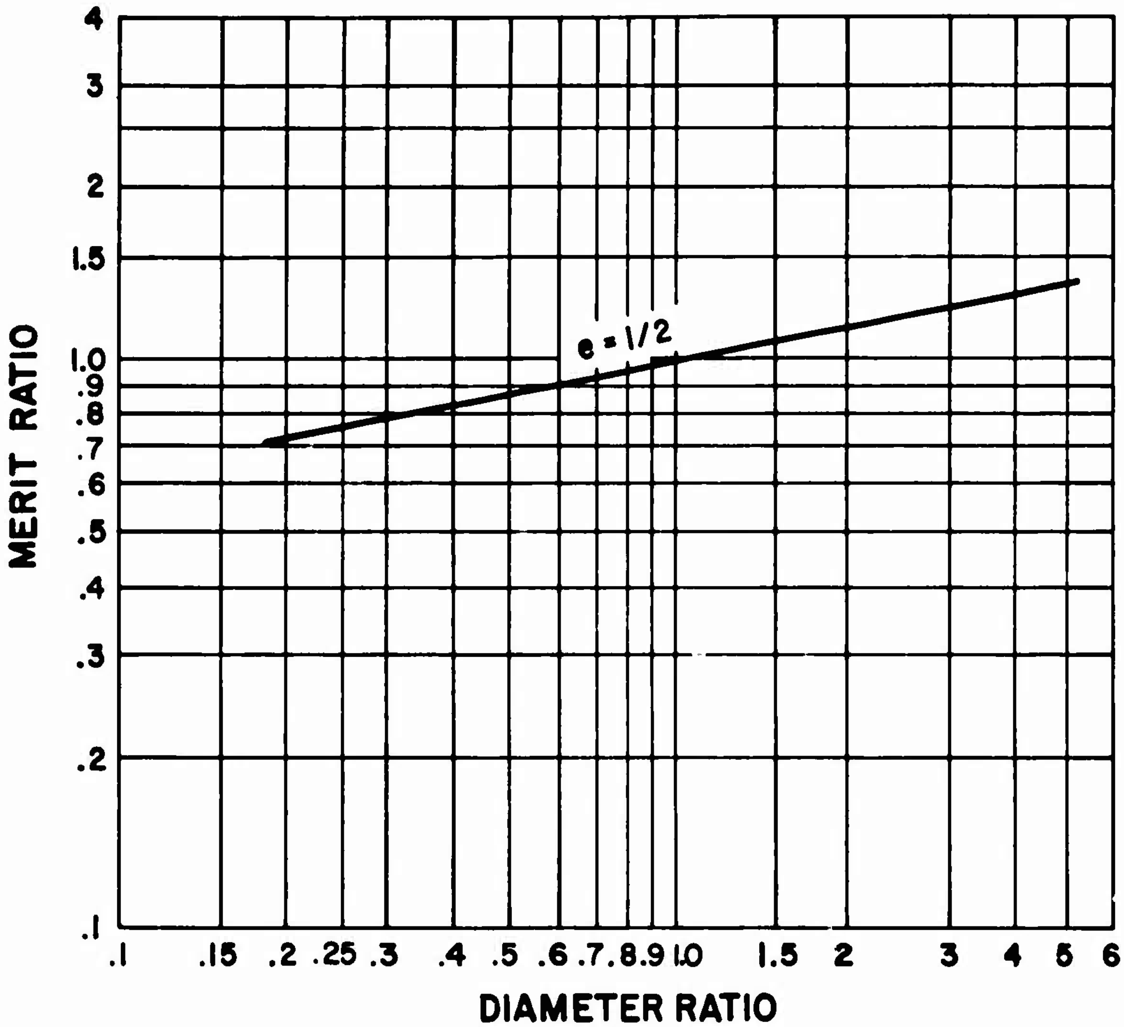

Figure 15. Merit Factor as Function of Diameter Ratio

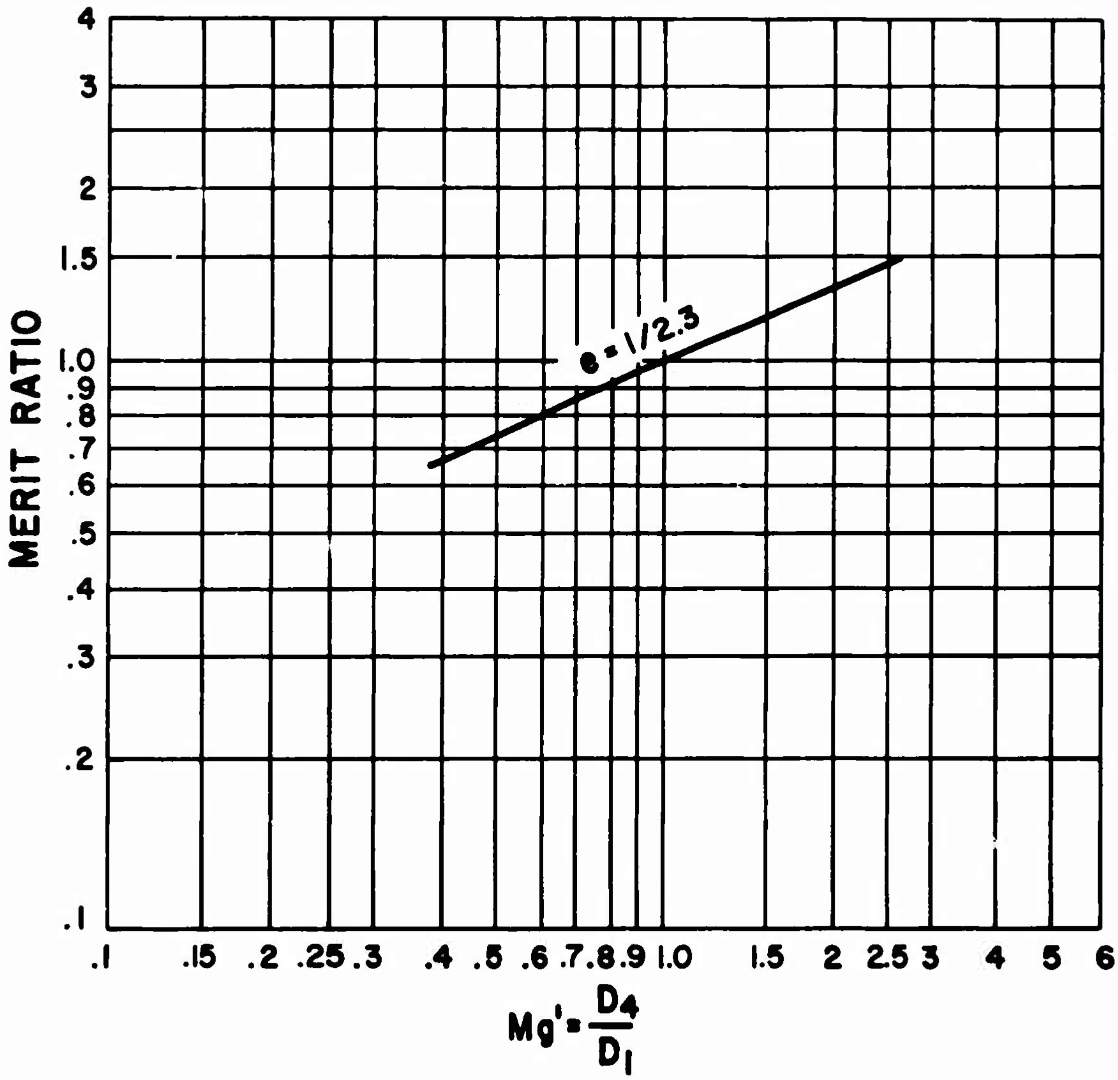

Figure 16. Merit Factor as Function of Primary-Secondary Ratio Spread

considered as being fairly elementary; future work along this line seems to hold out great possibilities of being productive to the general understanding of the operation and possible design trends to a much more sophisticated level than any effort to date. A few of the possible areas of refinement are mentioned in the following paragraphs.

Control of gear train tooth numbers by:

1. Providing hunting
2. Eliminating factoring
3. Eliminating tooth number factors over 89
4. Evaluate overall ratio error

Evaluate different types of bearing for:

1. Life-reliability factors
2. Total installed weight
3. Operating efficiency
4. Effects of replacement with journal bearings
5. Effect on failure

Refine the determination of tooth geometry by:

1. Include an exact determination of "J" factor
2. Optimize tooth thickness and addendum
3. Include involute modification for an appropriate tooth load decay

Include the application of helical gears.

Include the effects of series reliability elements.

Refine the determination of the merit factor:

1. Include current data as it becomes available
2. Add the effect of other significant factors
3. Broaden its scope to include other transmission types

Include an analysis of other elements:

1. Shafts
2. Couplings - flexible and fixed
3. Operational deflections
4. Bevel gear input sets

Consider transmission efficiency.

A parametric program for transmission design based on the current study and including these refinements would be a powerful tool for the development of transmissions to both helicopter and STOL installation.

PROTOTYPE TEST UNIT COSTS

An analysis has been conducted to determine the approximate costs to produce test prototype transmission of various horsepowers. These costs were separated into fixed tooling costs and hardware costs. Since these transmission types utilize standard machining methods and tolerances for all components, the prototype tooling consists mainly of patterns, forgings, dies, and possibly quenching dies. Also, since the machining of this type of transmission does not involve new techniques, the cost of fabrication can be estimated versus horsepower and torque based on previous units. These costs are shown in Figure 17.

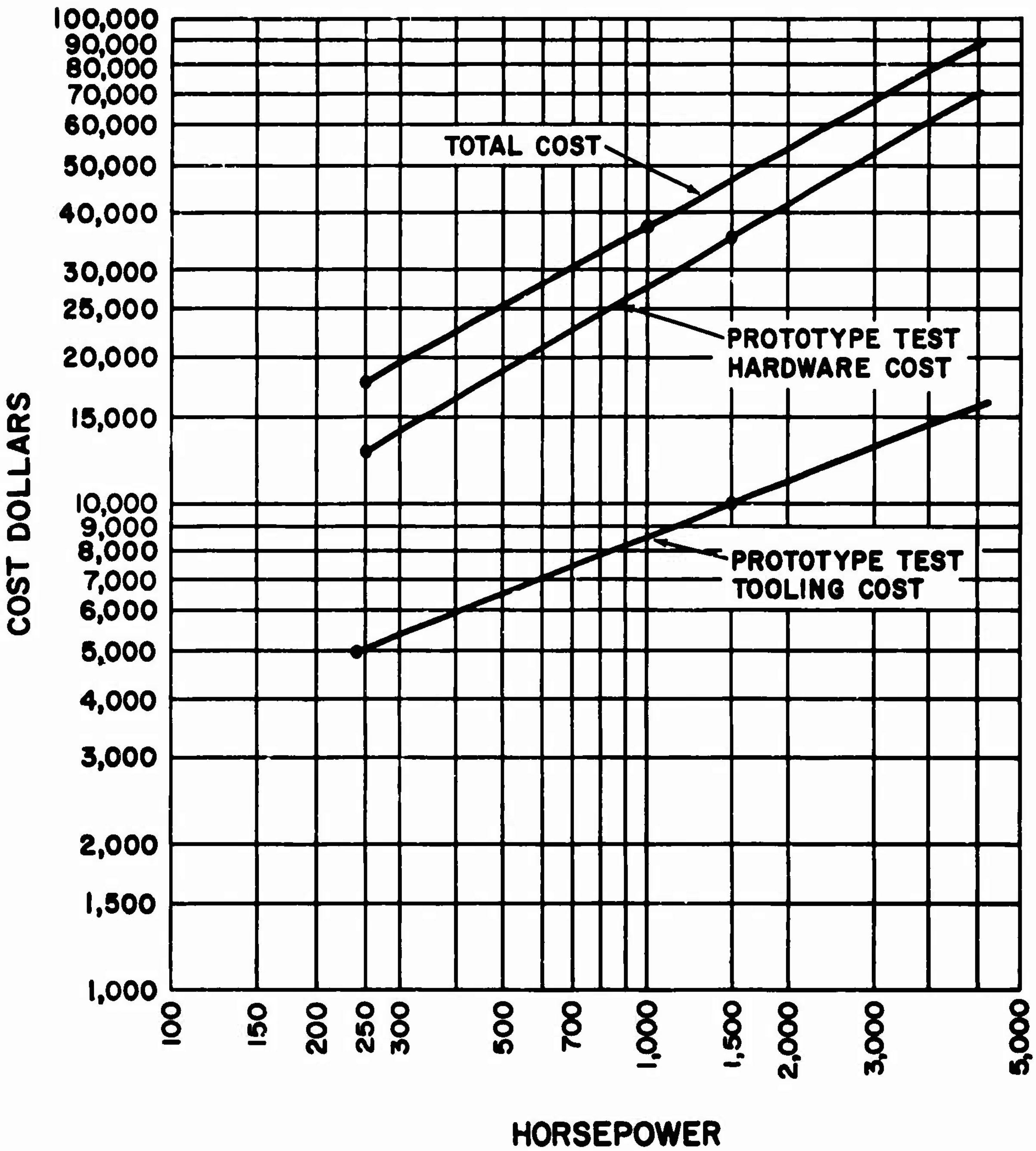

Figure 17. Prototype Test Unit Cases Vs. Horsepower

CONCLUSIONS

1. The compound planet system efficiently changes ratios with only two meshes in series, but since its design proportions become impractical above the 30:1 range it is feasible to consider in the ratios of 80 to 100:1 only when used in combination with an input reduction stage.

2. Handling over 95 percent of the torque and most of the speed reduction in the efficient two-mesh compound planet system can be combined with a conventional moderate ratio single-mesh input bevel gear stage to provide an advantageous net weight for the total three-mesh system.

3. The three-planet pinion Type III design is the most advantageous for Phase II hardware design consideration on the basis of weight and reduction in number of parts.

4. The load equalization capabilities of the low spring rate system studied permit a compound planet system utilizing standard manufacturing procedures which is capable of a 4000-hour system life with only a slight net installed weight difference.

5. The most sensitive planetary design factor is the space required to provide adequate planet bearing life since changing it affects the gear and carrier size significantly.

6. The many interrelated design parameters governing evaluation of total planetary design over a wide spectrum can be combined mathematically for efficient handling by means of computer programs.

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US Army Field Office, AFSC, Andrews AFB	1
Air Force Systems Command, Wright-Patterson AFB	1
Air Force Flight Dynamics Laboratory, Wright-Patterson AFB	1
Systems Engineering Group (RTD), Wright-Patterson AFB	3
Bureau of Naval Weapons, DN	7
Office of Naval Research	3
Chief of Naval Research	1
Bureau of Medicine and Surgery, DN	2
US Naval Air Station, Norfolk	1
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Commandant of the Marine Corps	1
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APPENDIX I

RATIO REQUIREMENTS - HELICOPTER TRANSMISSIONS

The ratio requirements of helicopter transmissions fall into two categories, each with a direct relationship to horsepower and output speed. The first category is the installation where the input to the transmission is directly coupled to the engine turbine wheel. The second category is the installation where the output speed of the turbine is reduced before it is coupled to the transmission.

Analysis of the present and projected engine designs indicated that the turbine speed varies inversely as the amount of developed horsepower increases (Figure 18). Further analysis indicated that engines with an integral stage of reduction normally provide their turboshaft power in the 6000 r.p.m. range (Figure 19). The requirements of rotor system r.p.m. versus horsepower are plotted (Figure 20).

This information is used to plot the ratio requirements versus horsepower. Mean-lines through these curves indicate that a ratio of approximately 87:1 satisfies all cases where the transmission is directly coupled to the turbine wheel. For all installations where engine speed reduction is provided, ratios in the range from 15:1 to 40:1 would be required depending on the horsepower. (Figures 21, 22, and 23 indicate these speed and torque relationships.)

Further study of helicopter transmission configuration requirements indicate that most rotor shafts will be at right angles to the centerline of the engine, requiring an input right angle gear stage. The use of an input stage is therefore, both necessary and convenient since it can readily provide 1:1 to 4.5:1 ratio reduction to the main transmission. Furthermore, accessory drives (hydraulic pump, tail rotors, alternators, et cetera) are always required and they are readily accommodated in the input set housing.

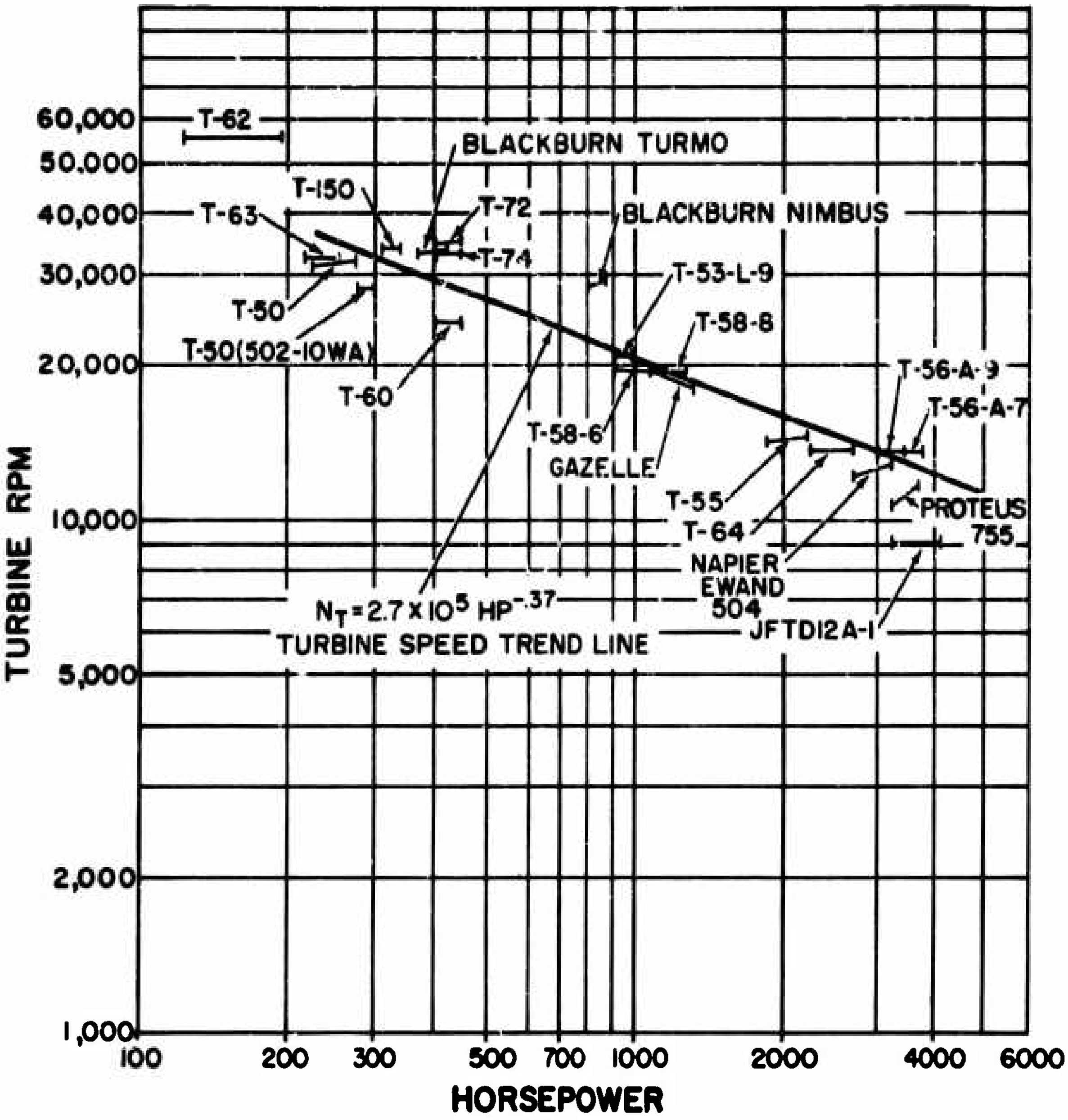

Figure 18. Gas Turbine Shaft Speeds (Full)

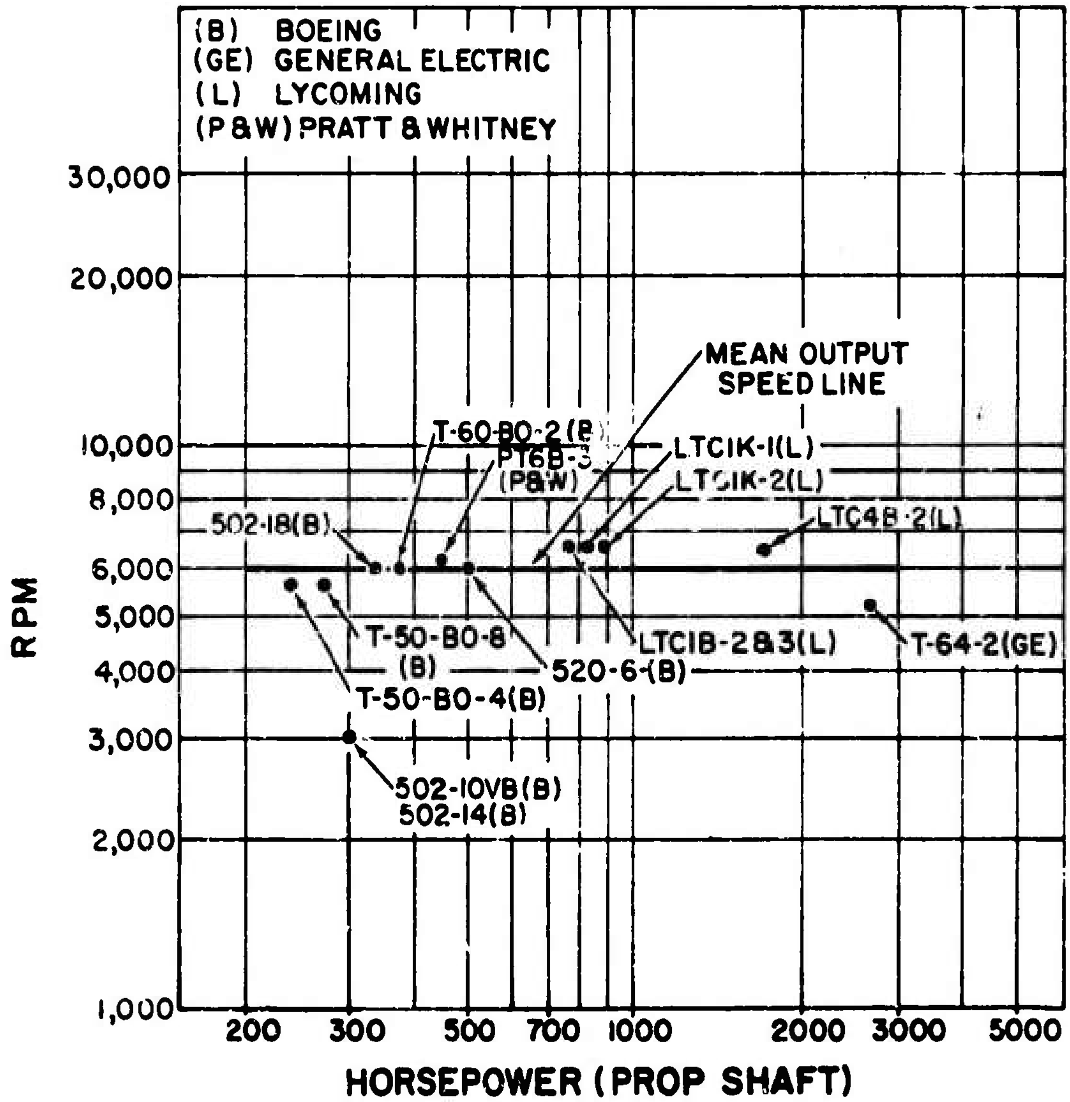

Figure 19. Gas Turbine Shaft Speeds Integral Reducation

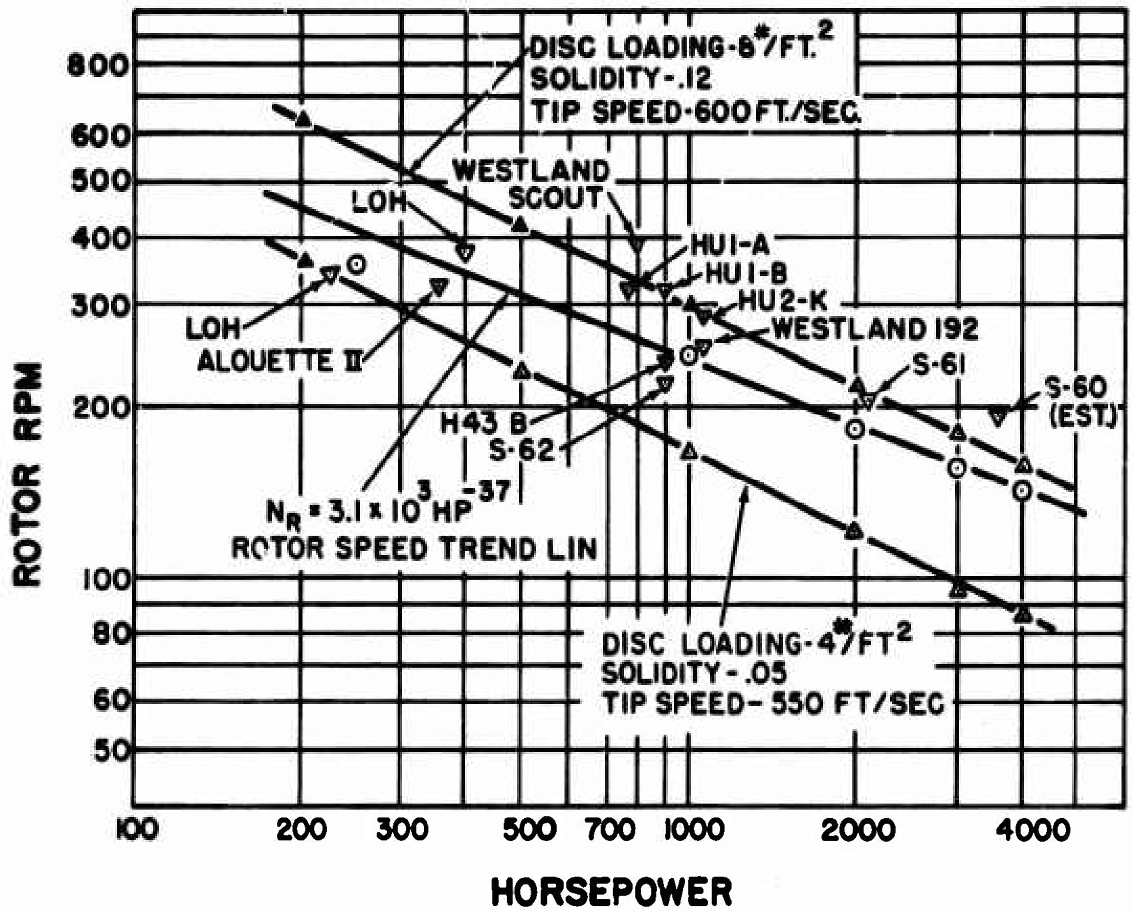

Figure 20. Helicopter Rotor Speeds

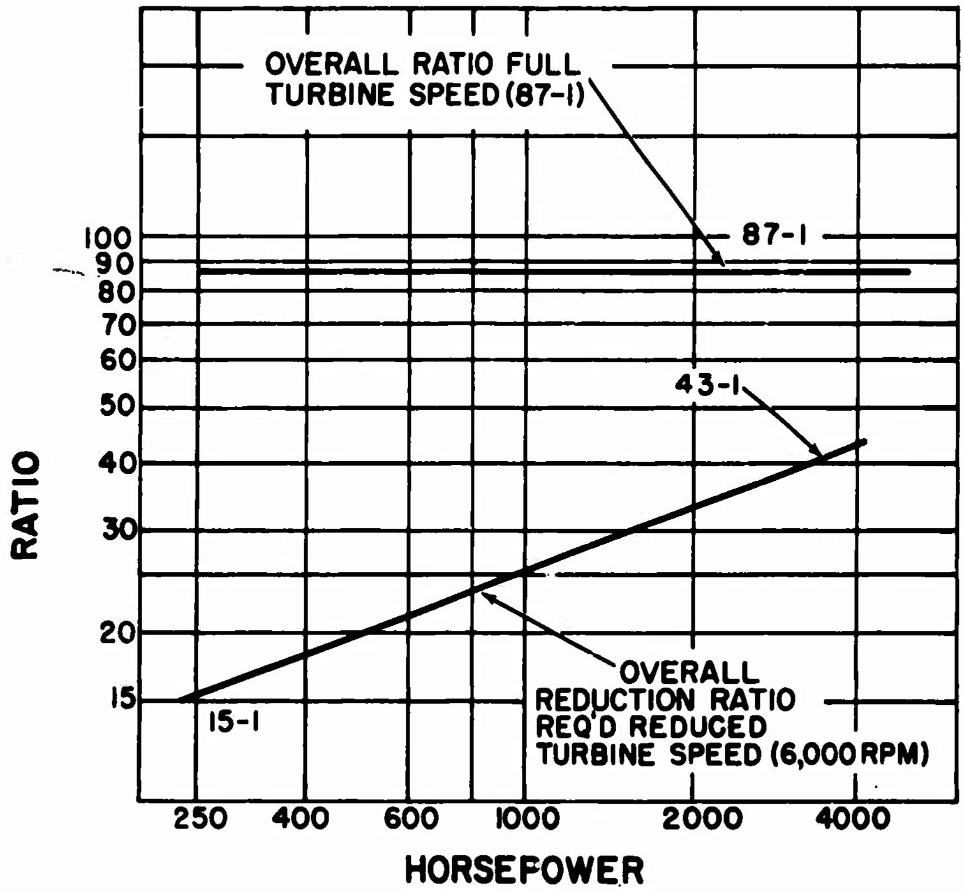

Figure 21. Transmission Ratio Requirements Vs. Horsepower

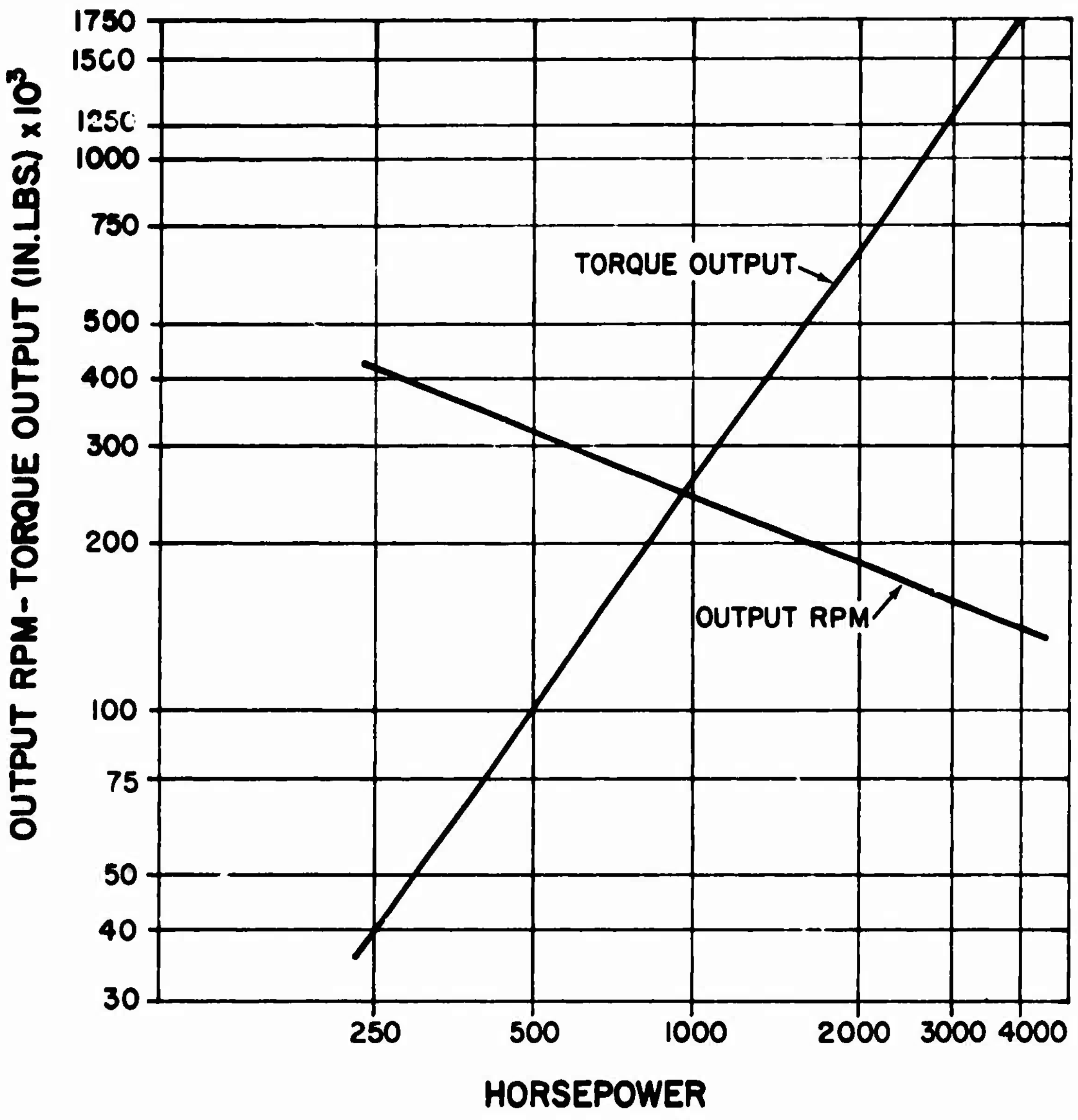

Figure 22. Horsepower-Output RPM and Torque Relationship

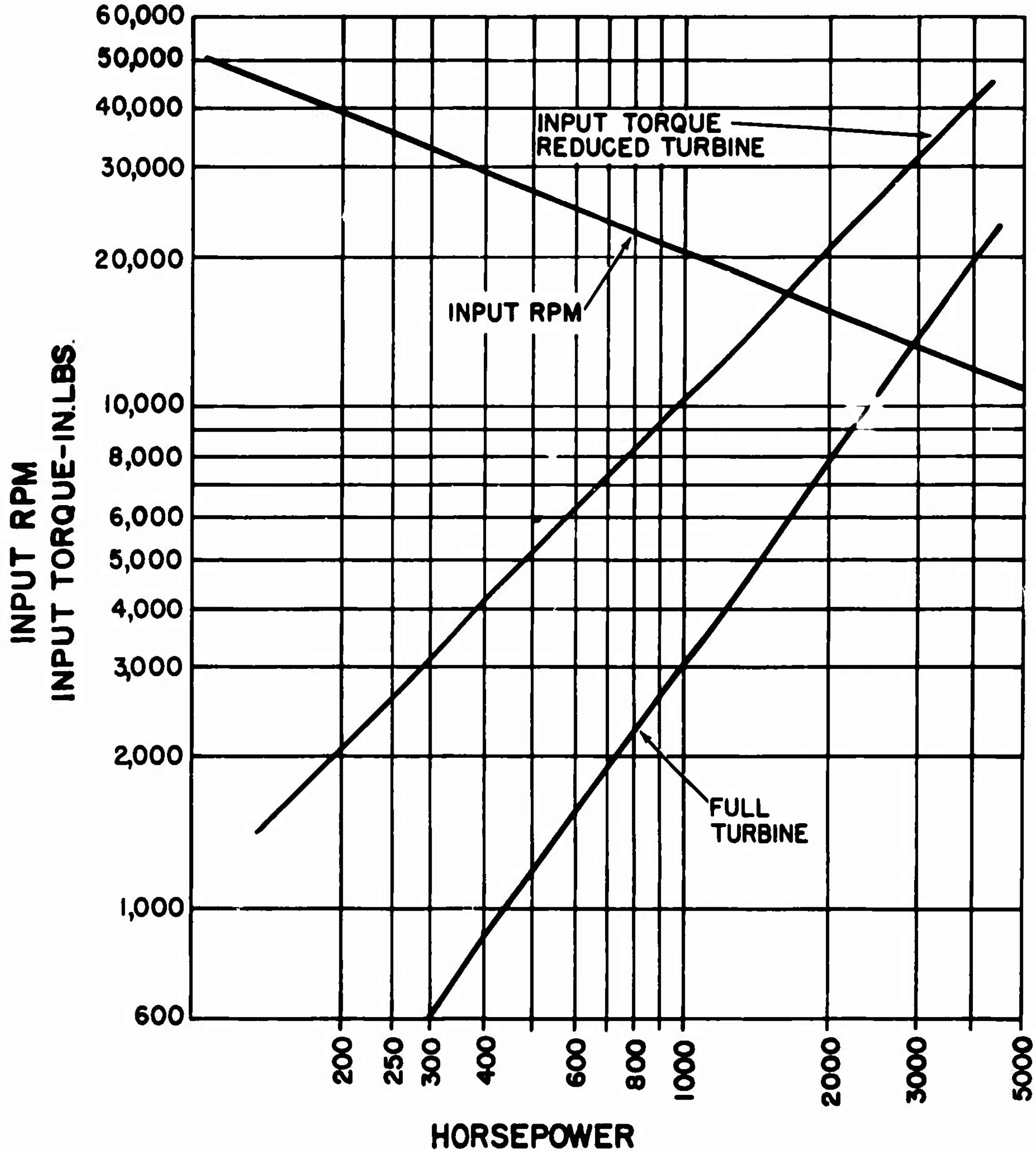

Figure 23. Horsepower-Input RPM and Torque Relationship

APPENDIX II

LIFE AND RELIABILITY CONSIDERATIONS

GEARS

The equations establishing gear design factors for life are presented in the Parametric Design Evaluation (pages 33 to 35) and are not repeated here. Certain design consequences arise from these equations. For instance, the size of the gear set

$$F_1 D_1^2 = \frac{166\, Q_1 \quad (G \;\omega_1 \quad L)^{0.1}}{H_b^2} \quad \frac{(1 + 1\,)}{mg_1}$$

Since for a given requirement torque (Q_1), and tooth hardness (H_b) are fixed, the variations in specific size depend upon load cycle frequency (W_1) and life (L_1) to the 0.1 power: For example, gears for 4000-hour life would be $\sqrt[10]{4} \cong 1.15$ times larger than those for 1000 hours. For the schematic design evaluations of Design Variation Evaluation (pages 11 to 21), therefore, we consistently used gear designs capable of 4000-hour life in all the design types and varied proportions only as required to satisfy bearing life requirements. In the computer study, life values for both gears and bearings were varied.

BEARINGS

Fatique life of the planet pinion bearings is the critical design variable. In considering bearing life and reliability it is necessary to define the assumptions one uses since designs are valid only when compared to the same life standards. Our work is based upon using manufacturer's B_{10} life values for rolling contact bearings as the fundamental information. This information is universally used and available. Furthermore, we apply this information to govern design choices in two major ways as follows:

1. Design Life of Individual Components

   For many years designs have been proposed and represented as having, for example, an expected life of 1000 hours when the individual B_{10} life rating of the bearings was 1000 hours. This is not statistically or practically correct but in the case of designs

with the same number of working bearings the design lives are relatively comparable.

2. Design Life of Systems

It is a requirement in most military hardware for the system to operate satisfactorily for some reasonable period in order to minimize maintenance costs and assure completion of assigned missions. The governing design rules must account for the reliability of a number of individual elements working in parallel and series arrangements as a system. As of this writing, there is no universally accepted andused basis for mechanical system life prediction. It is, therefore, necessary to make assumptions which will not be identical with other designers concepts. This makes valid comparison of competitive design proposals impossible. A gear train with 6 to 12 life-limited planet pinion bearings is extraordinarily large and heavy when 5 to 10 times the bearing life capacity is provided. When, conversely, individual component life is assumed for the system, the system design life is not adequate. These two extreme assumptions may be resolved by combining experience and theory. Experience shows that bearings in this type of application will operate longer than the catalogue B_{10} rating established by the manufacturer. By combining this knowledge with the established theory of system life, a more realistic basis for design is applied throughout this parametric design study.

We assume the following:

Improved overall system life is required; therefore, our analysis must include allowance for the effect of number of working parts on system reliability.

Value in Table 3 represents catalogue B_{10} life required to get proper life using the graph defining the L_s/L_c system reliability relationship.

TABLE III SYSTEM BEARING RELIABILITY CONSIDERATIONS				
	Individual Basis B_{10} 1000 Hours	System Basis B_{10} 2000 Hours	System Basis B_{10} 3000 Hours	System Basis B_{10} 4000 Hours
1 BEARING	1000	400	600	800
6 BEARINGS	1000	2220	3340	4450
12 BEARINGS	1000	4450	6670	8900

It is feasible to make a first order determination of the most economical design life for a military transmission or other life limited component. This has been done using data from published minutes of Congressional hearings (6) and certain other data as shown on the following page. This indicates that for the FY 61 helicopter maintenance totaled $75,000,000 for U.S. Army vehicles; which amounts to about $7,500 per year for each operational transmission.

An ultimate goal can be defined in terms of an increase in TBO to the total expected flight time to the obsolescence of the vehicle. This would essentially eliminate any TBO restriction and would also eliminate all scheduled maintenance costs. Further, vehicle utilization would increase so that total flight activity could be handled with fewer vehicles. An estimate of the factors that can be obtained from such an optimization have been determined. This indicates that helicopters having a TBO of 4,000 hours could save in transmission maintenance only $18,500,000 per year.

Ref: (6) O&M Section, FY 61 Minutes of Hearing of House Appropriations-Armed Forces Subcommittee

ECONOMIC TRANSMISSION DESIGN LIFE

Let:

C = cost of procurement and maintenance/year
X = cost of maintenance/year
G = number of years per generation
t = TBO hours
n = number of overhauls per generation
μ = hours flight/hours available utilization factor
N = number of helicopters
2000 = number of hours nominally available/year

Since:

$C-X$ = procurement cost
2000μ = yearly flight time-hours
$n = 2000 \frac{\mu}{t} G-1$

Present status: Ref: (7)

$G = 8$
$X/C = 0.5$
$\mu = 0.25$
$t = 400$
$C = 150 \times 10^6$
$2000\,\mu = 500$ hour/year/helicopter
$N = 2500$
$2000\mu N = 500 \times 2500 = 1{,}250{,}000$ flight hours/year

$\frac{X}{N} = \frac{75 \times 10^6}{2.5 \times 10^3} \cong \$30{,}000$ overhaul cost/helicopter/year

$\frac{\$30{,}000}{4} = \frac{\$7{,}500 \text{ overhaul/transmission}}{\text{year}}$

Assume an ultimate goal:

$X = 0$
$\mu = 0.25$
$G = 8$

Therefore:

$2000\,\mu = 500$
$t = 2000\mu G = 4000$ hours (required TBO)
$2000\mu N = 1{,}250{,}000$ hours/year (assumed operational level)

Then:

2500 helicopters with transmission overhaul eliminated

Savings = 2500 x \$7,500 = \$18,500,000/year

Ref: (7) Overhaul and Maintenance Section, FY 61 Congressional hearing pages 72, 74, 75, and 87

The actual life expectancy of bearings is appreciably better than B_{10} values given in the catalogues.

A reasonable design goal is 4000-hour system life expectancy.

These assumptions have been applied as shown in the accompanying graph (Figure 24), and Tabulation (Table III). The graph provides a plot of overall system reliability at a chosen system life (L_s) as it varies with number (n) of bearings in the system. This is given in dimensionless form as the ratio L_s/L_c the ratio of the desired system life (L_s) to the individual bearing B_{10} catalogue life rating (L_c). This design factor is predicated upon the basis of the actual individual bearing life rating being closer to B_2 in practice than B_{10} value given in catalogue ratings.

Based upon reliability tests conducted by this contractor for the Bell Telephone Laboratories (8) and based upon bearing failure analysis data published by the General Motors (9), the use of the L_{B2} assumption leads to good design proportions and reliability. Bearing selection by this system is summarized for several typical requirements in the tabulation following the L_s/L_c Chart.

Ref: (8) Rezeau-SAE Journal September 1957
(9) Timken Report No. 151.3-G

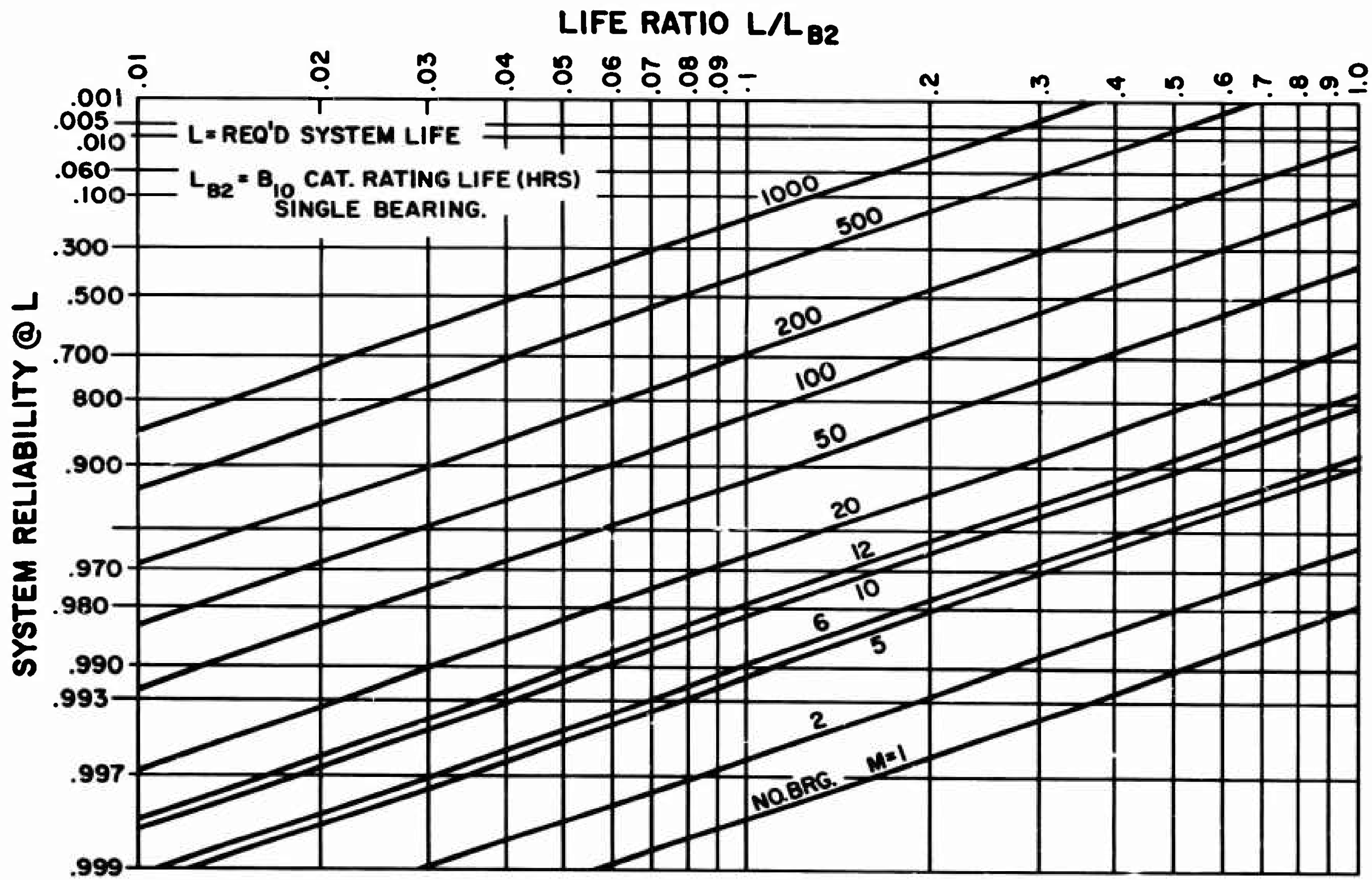

Figure 24. System Life Vs. Individual Life

APPENDIX III

GENERALIZED TRANSMISSION DESIGN DATA

In choosing design factors for gears and bearings suitable for all powers and ratios, it is convenient to reduce the relationships as much as possible so that extrapolation from one design to another is simplified. The most general rule adopted which is helpful in this respect is as follows:

$$\text{Size of design}_2 = \text{design}_1 \sqrt[3]{\text{torque}_2/\text{torque}_1}$$

Varying the size of a design varies the torque capacity directly as the cube of any dimension. For example, a design 1.26 larger has 2 times the torque capacity. Proof of this general rule as it applies to several functions is:

Gear Compressive Stress (Wear) Factors

At the same gear ratio (mg), life (L), number load cycles (w), and hardness of teeth H_b vary only the torque and check the variation in size required.

From gear data $F_1 D_1^2 = KT_1$

Assume $T_2 = 8T_1$

then $D_2 = D_1 \sqrt[3]{T_2/T_1}$

where

*F_1 is the face width
*D_1 is the diameter
T is the torque
K is the constant (includes mg, L, w, etc.)

*Assume that proportions of F to D remain constant, that is, design is photographically enlarged.

Gear Strength

@ same ratio, life, load cycles, hardness, et cetera

From gear
strength data $N_1 = KY_K'$

where

N_1 is the number teeth
Y_K is the tooth strength factor

then $N_1/2D_1 \; D_2 = 2D_2$

the tooth is approximately twice as strong at 1/2 diametrical pitch. From previous $T_2 = 8T_1$ but $F_2 = 2F_1$ @ doubled design size tooth load is only 4 times as large since diameter is doubled, tooth load per face width is only twice since face width is doubled. Load $_2/F_2$ is twice and tooth form is twice as strong so the tooth strength stays constant when the tooth numbers are held constant and the design size is increased in relation to torque as previously proven.

Bearing Life Factor (Planetary)

$$P = \text{pinion bearing load}$$

$$P = \frac{T}{NR_T}$$

$$\text{volume brg.} = D_B^{\,3}$$

and $D_B^{\,3} = KC^{1.5}$
@ constant C/P and speed brg. life is constant

where

T is the torque
N is the number of pinions
*R_T is the radius to pinion bearings
*D_B is the diameter bearing
P is the bearing load

*Proportions of R_T & D_B are constant

then: $$R_{T_2} = R_{T_1} \sqrt[3]{T_2/T_1}$$

This confirms the rule that proportional change in design according to cube root of torque variance will keep bearing life factors constant.

The foregoing illustrates variation in gear and bearing size to keep pace with a different torque requirement. For the case where the gear design is sized correctly for a specific torque, it is generally found that further size variation, if any, is necessary to accommodate desired life factors for bearings. This involves change in design size at the same torque in order to change life

for example $$L = \text{bearing life} = K\left(\frac{C}{P}\right)^{10/3}$$
$$D_B^{\,2} = KC$$
$$P = \frac{K}{D_B}$$

when $T = K$ & R_T increases with D_B proportionally

then: $$D_{B_2} = D_{B_1} \sqrt[10]{L_2/L_1}$$

For case where only bearing diameter increases and transmission dimension R_T stays constant, then at constant torque

$$P = \text{constant}$$
$$L = K\left(\frac{D_B^{\,2}}{P}\right)^{10/3}$$

then: $$D_{B_2} = D_{B_1} \sqrt[6.66]{\frac{L_2}{L_1}}$$

For helicopter transmissions, the speeds of planet pinions are slow enough to permit use of needle proportioned roller bearings. These bearings generally permit the greatest life per unit volume or weight, all factors considered. The range of available commercial heavy duty roller bearings, for example, bear out the assumption that bearing weight, volume, and diameter^3 are proportional to bearing dynamic load capacity $^{1.5}$.

Designs Type I and II use six interleafed planet pinions driven by a dual sun gear arranged to exactly divide the load. The design parameters for these configurations, based upon stress limitations in the rocking lever system, are presented in Figure 25. This establishes minimum sun gear sizes in establishing the designs for comparative weight evaluation. The three basic stresses determining the minimum size of the rocking lever system are quill-shaft stress, ball joint driving stress, and ball joint bearing pressure. Analysis prepared in the development of a 4000-horsepower turbo-prop transmission has indicated an acceptable design level for these stresses. This illustrates the minimum dual sun gear pitch diameter required for a given torque.

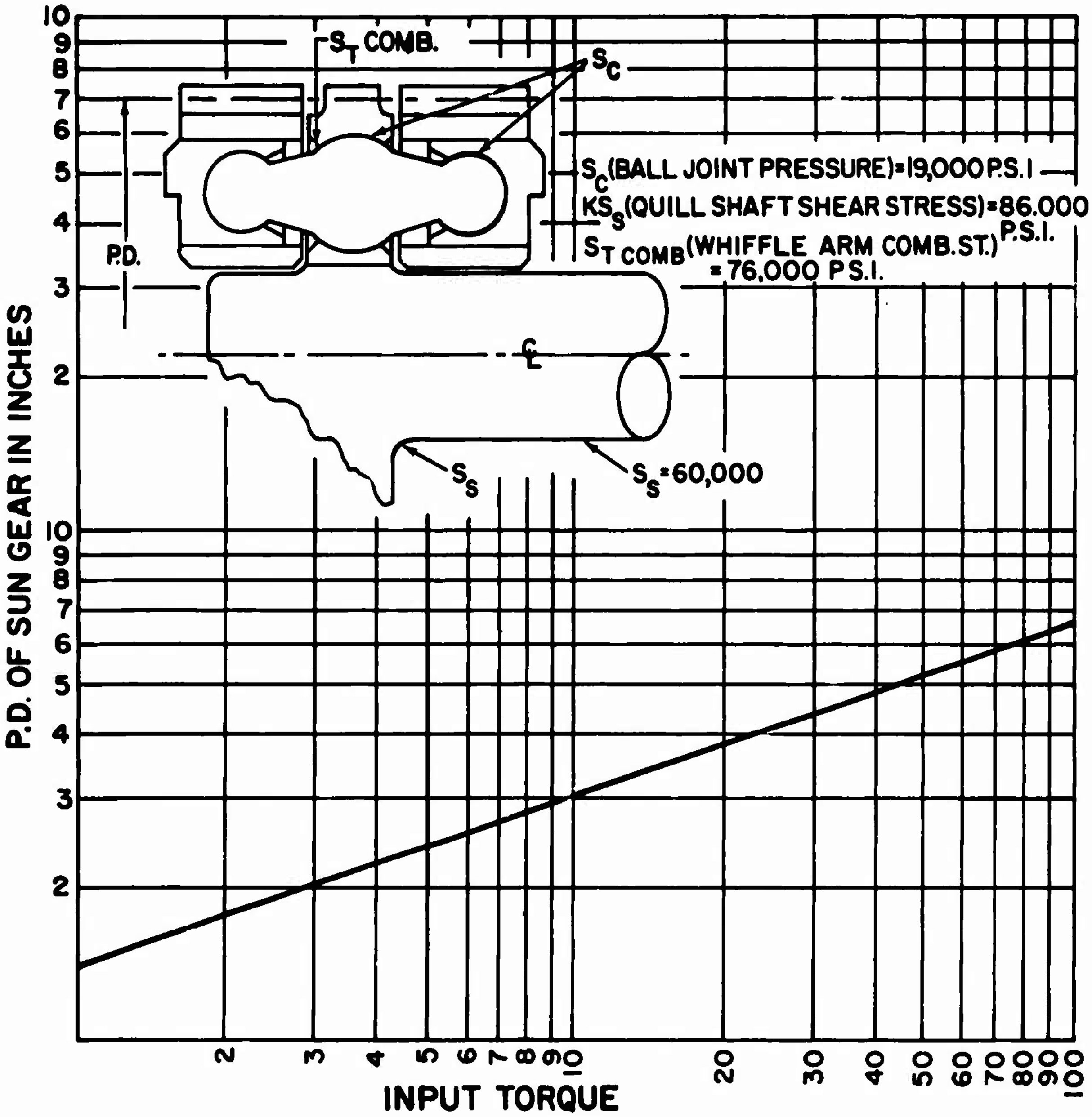

Figure 25. Minimum Whiffletree Diameter Requirements Vs. Input Torque

APPENDIX IV

TRANSMISSION EFFICIENCY

One measure of the operational efficiency of a helicopter is the ratio of useful weight to gross weight. Every pound of unnecessary component weight saved means an added pound of useful load. With the advent of the gas turbine engine, considerable weight saving in the propulsion system of the helicopter is realized. However, the higher output speed of this type of engine requires greater overall speed reduction with some increase in the weight of the power transmission system. Since the lifting capability of a helicopter depends on the power transmitted to the rotor, the ratio of rotor power to available engine power as determined by transmission efficiency is an important factor.

The primary variables affecting power transmission efficiency are number of gear meshes, degree of power recirculation, specific mesh efficiency, and bearing efficiency. The compound planet system, herein considered, provides 15 to 30:1 ratio change with only two-series meshes and with no power recirculation. The conventional simple planetary requires four and possibly six-series meshes for the same reduction. Since, as shown later, the power loss per mesh is of the order of 0.5 percent to 0.8 percent, the extra power supplied to the rotor with two instead of four meshes can easily be 1-percent to 6-percent greater.

The power loss expected with this gearing arrangement is 15.85 horsepower out of 1000-horsepower input of approximately 98.4 percent efficiency in the main reduction unit. The power loss at each gear mesh is estimated on the basis of Figure 26 which gives a good first approximation based upon the main influencing variables of gear ratio, internal or external mesh, and number of teeth in the pinion.

Differential means for achieving a high reduction ratio in a few meshes will obtain recirculation of between 2 and 5 times the input power at the bearings and gears respectively. The individual bearing gear efficiencies are high, but due to power recirculation, the entire power loss is increased to the order of 5.2 percent.

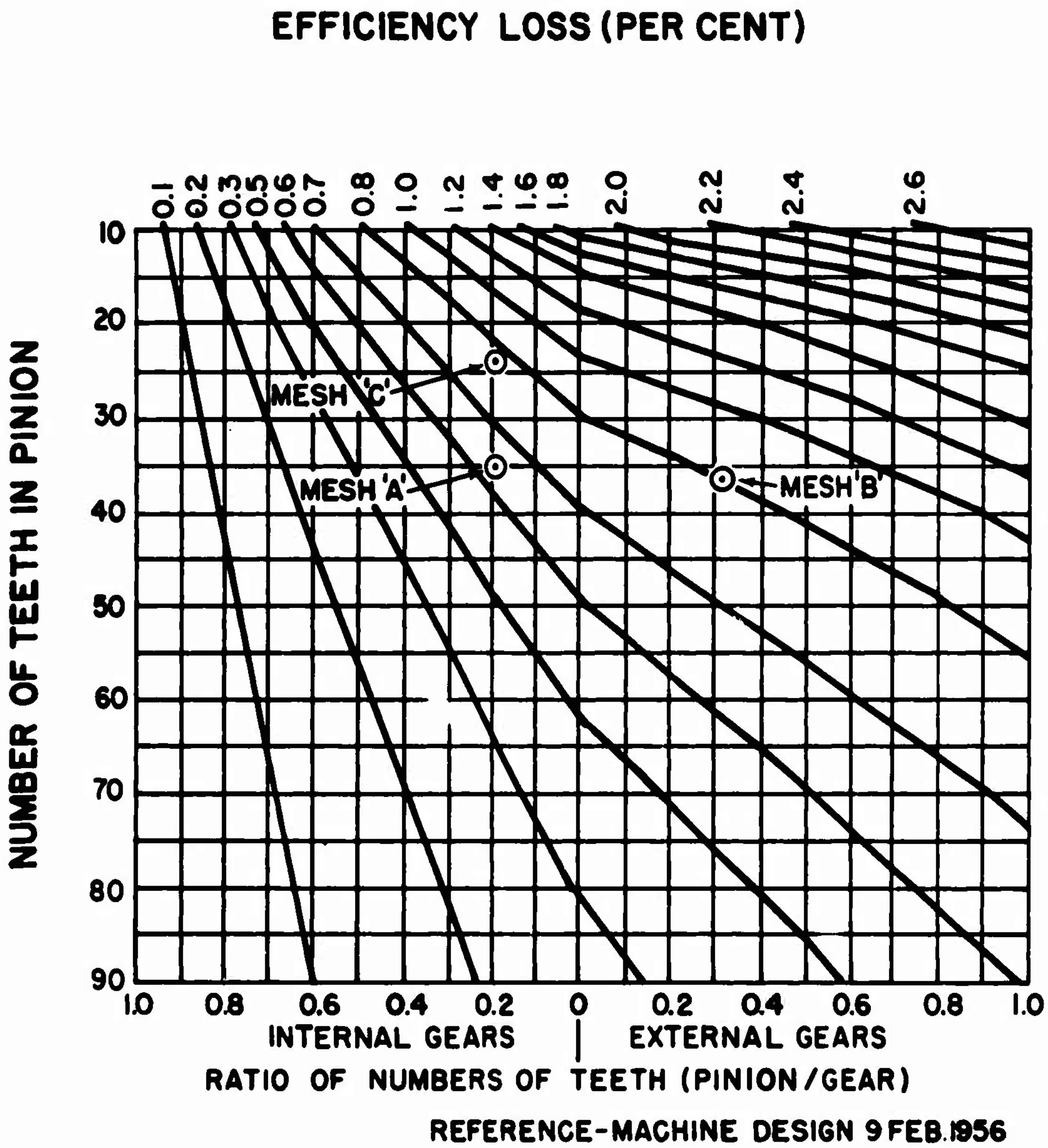

Figure 26. Efficiency Loss (Percent)

The power loss in the transmission is quite significant. Although individual mesh efficiency is not subject to appreciable control by the transmission designer, important advantages are gained by using configurations which minimize the required number of series meshes without power recirculation. The compound planet system meets this criterion. Consider, for example, two helicopters identical except for the transmission efficiencies:

Transmission weight	470	470
Transmission efficiency	98.4	94.8
Engine power to transmission	1000	1000
Power to rotor	984	948
Gross lift	9840	9480
Empty weight	6500	6500
Net useful load	3340	2980

In the case illustrated, the more efficient transmission permits lifting 360 pounds additional useful load -- a significant 12 percent increase. The example given is based on the assumption, borne out in our studies, that the compound planet system even at increased TBO values, can be the same weight or less than the equivalent conventional doubled simple planetary arrangement.

A more efficient transmission can add weight to the design to achieve other goals and still have a net weight advantage over the less efficient transmission. For example, 70.5 pounds (15 percent of 470 pounds) added to the life limited features could readily increase the expected TBO by a factor of 3 or 5 times and still have a net weight advantage in the example given of 290 pounds.

The design for Phase II of this contract was finalized at a life factor assuming an expected TBO of 4000 hours. This was done within existing state-of-the-art weight values. Less weight could have been obtained at shorter life but 4000 hours appears to be the most economical TBO. Since this Phase II design is also of the highest efficiency, the combination of long TBO weight efficiency, and competitive design weight makes the overall helicopter installation most advantageous.

Calculations are included to illustrate a specific example of the high efficiency possible (97.7 percent) with this design:

The power transmitted in terms of load and velocity is figured for each of the meshes and bearings of a typical compound planet transmission. The loss at each station in the sequence of power flow from input to output is figured to arrive at the total power loss and overall efficiency.

Input power: 1000 horsepower @23,9000 r.p.m. and 2630 inch-pounds torque

All notations and headings refer to Figures 26, 27, and 28.

Mesh A (auxiliary mesh)

$$P_A = \frac{\text{torque}}{R_G} = \frac{2630}{1} = 2630 \text{ pounds}$$

$$V_A = \frac{\text{RPM } (\pi) \quad (D_G)}{12 \quad (60)} = \frac{(23{,}900)\ (3.14)\ (2)}{720}$$

$$= 209.6 \text{ feet/second}$$

$$P_A\ V_A = 2630 \quad (209.6) = 550{,}000 \text{ ft.lb./second}$$

$$HP = \frac{550{,}000}{550} = 1000$$

Gear loss eff. percent from Figure 26 = .064 percent
Loss @mesh A = gear loss eff. percent (HP) = .064 percent (1000) = 6.4 horsepower

Bearing A_1

$$P_{A1} = 2630 \text{ pounds (bearing reaction)}$$

$$V_{A1} = \frac{23900\ (\pi)\ \ (.75)}{720} = 78.2 \text{ feet/second}$$

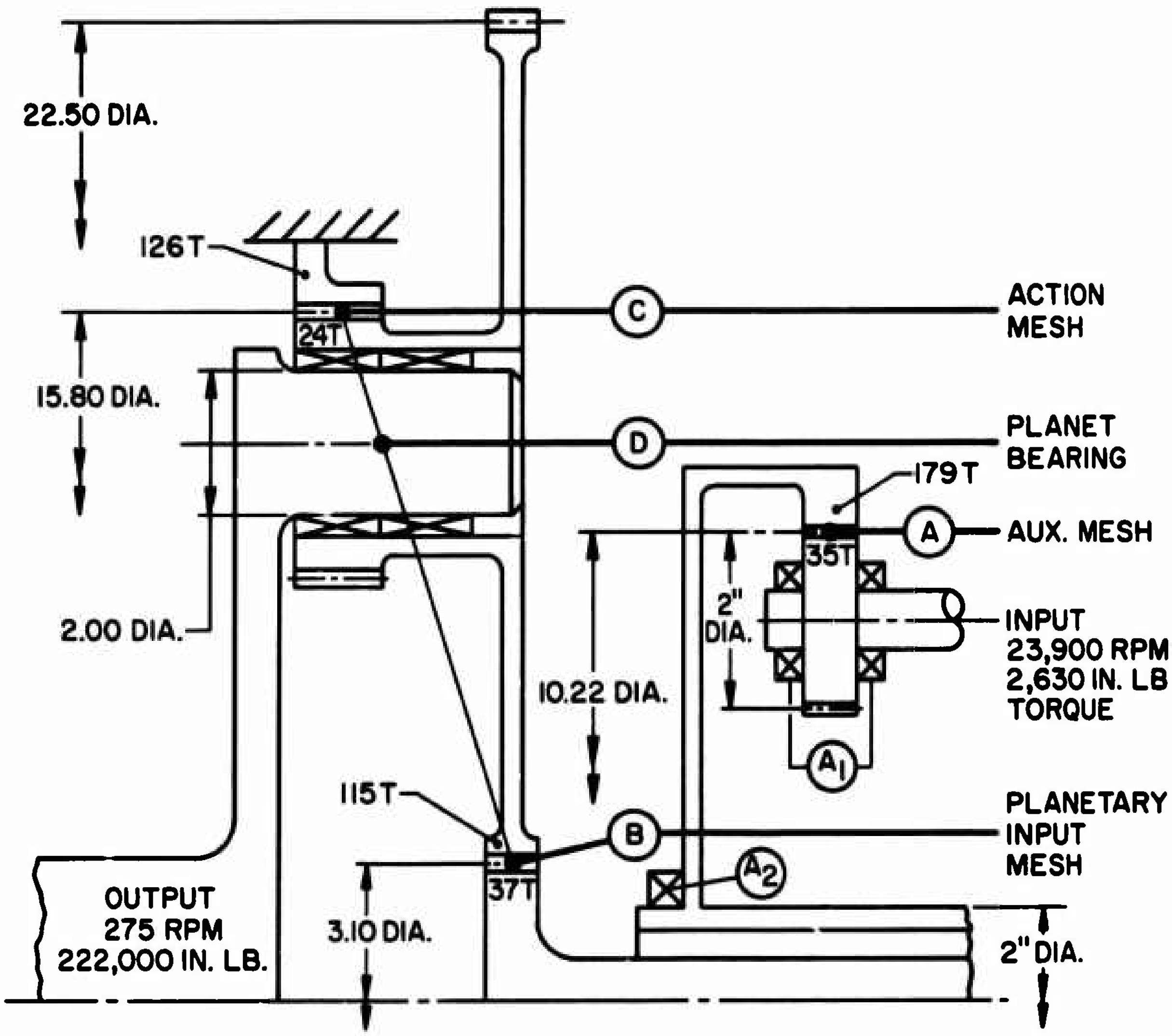

Figure 27. Three-Mesh Nonregenerative Compound Planetary - 87:1 Total Reduction

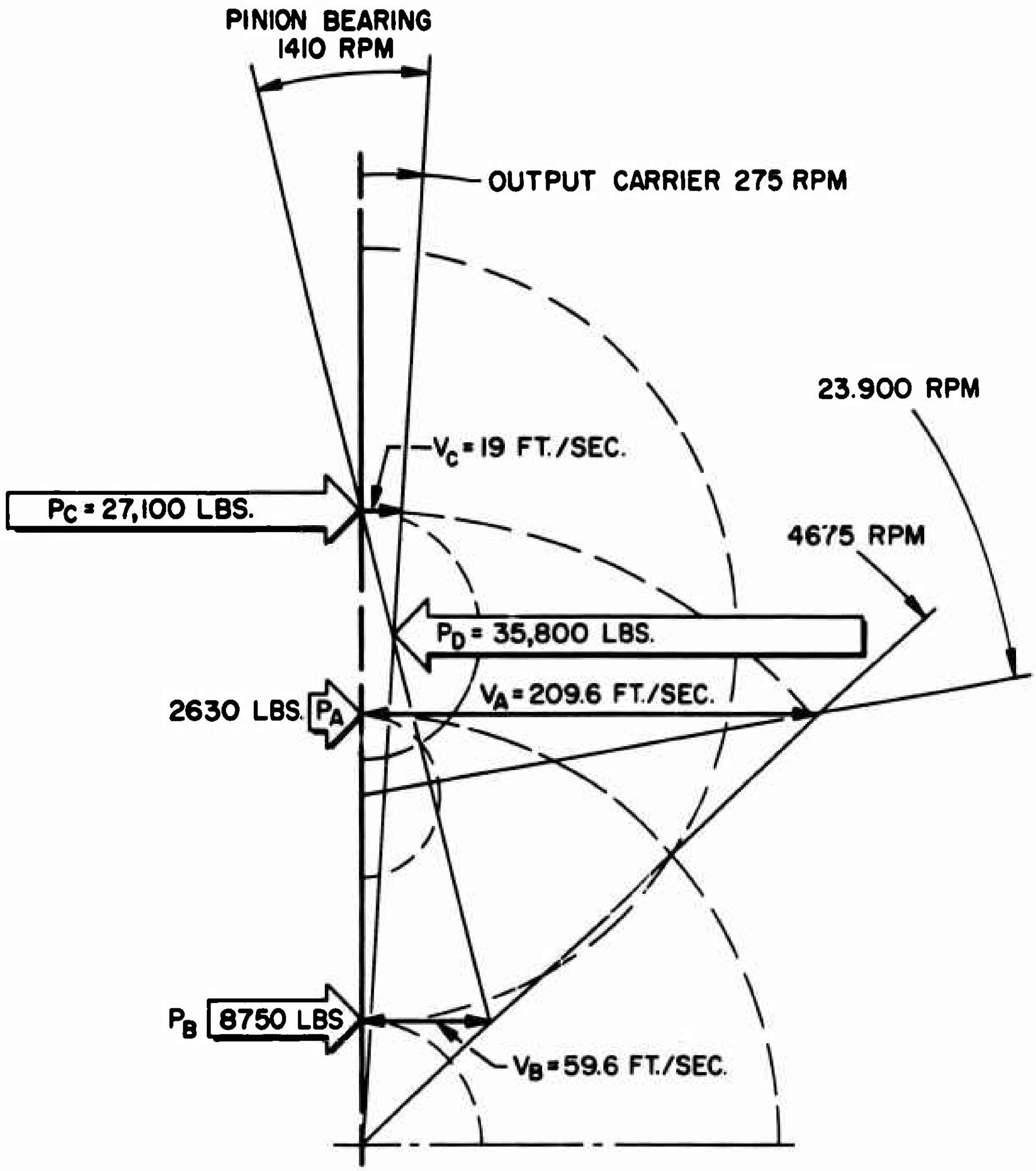

Figure 28. Instantaneous Velocity

$P_{A1}V_{A1}$ = 2630 (8712) = 206,000 feet-pounds/second

$$HP = \frac{206{,}000}{550} = 374$$

Loss @ brg. = bearing loss efficiency percent horsepower equals *0.15 percent (374)

Loss @ brg. = 0.55 horsepower

Bearing A_2

P_{A2} = 2630 pounds (bearing reaction)

$$V_{A2} = \frac{23{,}900\ (35/179)\ (2)\ (\pi)}{720} = 40.8 \text{ feet/second}$$

$P_{A2}\ V_{A2}$ = 2630 (40.8) = 107,500 feet-pounds/second

$$HP = \frac{107{,}500}{550} = 195$$

Loss @ brg. = bearing loss efficiency percent horsepower equals *0.15 percent (195)

Loss @ brg. = 0.293 horsepower

Mesh B (planetary input)

$$P_B = \frac{(2650)\ (179/35)}{1.55} = 8750 \text{ pounds}$$

$$V_B = \frac{[(23900)\ (35/179) - 275]\ \pi\ (3.10)}{720} = 59.6 \text{ feet/second}$$

$P_B\ V_B$ = 8750 (59.6) = 516,000 feet-pounds/second

HP = 940 horsepower

Gear loss efficiency percent from 9-3 equals 0.8 percent

Loss @ mesh equals gear loss efficiency percent horsepower (.993) equals .8 percent (.993) (940)

*Bearing loss efficiency % = 0.15 percent (SKF and others)

Mesh C (reaction mesh)

$$P_C = \frac{\text{torque output - planetary torque input}}{R_G \text{ (ring gear)}}$$

$$= \frac{229{,}000-13{,}490}{-7.95}$$

P_C = 27,100 pounds

$$V_C = \frac{275 \ (\pi) \ (15.8)}{720} = 19 \text{ feet/second}$$

$P_C \; V_C$ = 27,100 (19) = 516,000 feet-pounds/second

HP = 940 horsepower

Bearing loss efficiency percent from Figure 30 equals 0.74 percent

Loss @ mesh equals gear loss efficiency percent horsepower (***.9853) equals .74 percent (.9853) (940)

Bearing D (planet bearing)

$$P_D = \frac{229{,}000}{6.38} = 35{,}800 \text{ pounds}$$

$$V_D = \frac{1410 \ (\pi) \ (2)}{720} = 12.36 \text{ feet/second}$$

$P_D \; V_D$ = 35,000 (12.36) = 443,000

HP = 800 horsepower

Loss @ brg. equals bearing loss efficiency percent horsepower (***.9853) equals (*.15 percent) (800) (.986)

Loss @ brg. eauals 1.19 horsepower

*Bearing loss efficiency percent equals 0.15 percent (SKF and others)

***Horsepower correction factor to allow for losses in planetary input stage and auxiliary stage

Total losses equal $\sum \text{mesh } A + B + C + \sum \text{brg. } A_1 + A_2 + D$

Total losses equal $[6.40 + 7.46 + 6.85] + [.55 + .293 + 1.19]$

Total losses equal $\left[\begin{matrix}20.71\\ \text{gears}\end{matrix}\right] + \left[\begin{matrix}2.033\\ \text{bearings}\end{matrix}\right]$

Total losses equal 22.74 horsepower

Total efficiency losses equal 2.27 percent

Overall system efficiency equal 97.73 percent

Net system output - 275 r.p.m., and 222,000 inch-pounds
977.3 horsepower

Overall ratio - 87:1

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13. ABSTRACT
This report presents a complete review of the parametric design and computer studies performed to evaluate the Bergen Research Compound Planetary Transmission concept in three configurations for a helicopter transmission as schematically illustrated in Figure 1. All three concepts embody design arrangements which assure uniform load division with multiple compound planetary gears. Uniform loading is accomplished by balancing the primary and secondary meshing point positions through low rate springing. These systems stabilize at the gear mesh positions of load path equalization. Design are established for typical power ranges using consistent gear and bearing design factors. Both 1000 and 4000 hour time before overhaul design factors are studied to determine their feasibility and design trade off requirements. The Phase I study results are applied in Phase II to the preferred design concept through completion of a specific layout and detail design. This 250 Horsepower/ 4000 Hour design is suitable for fabrication and life testing.

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